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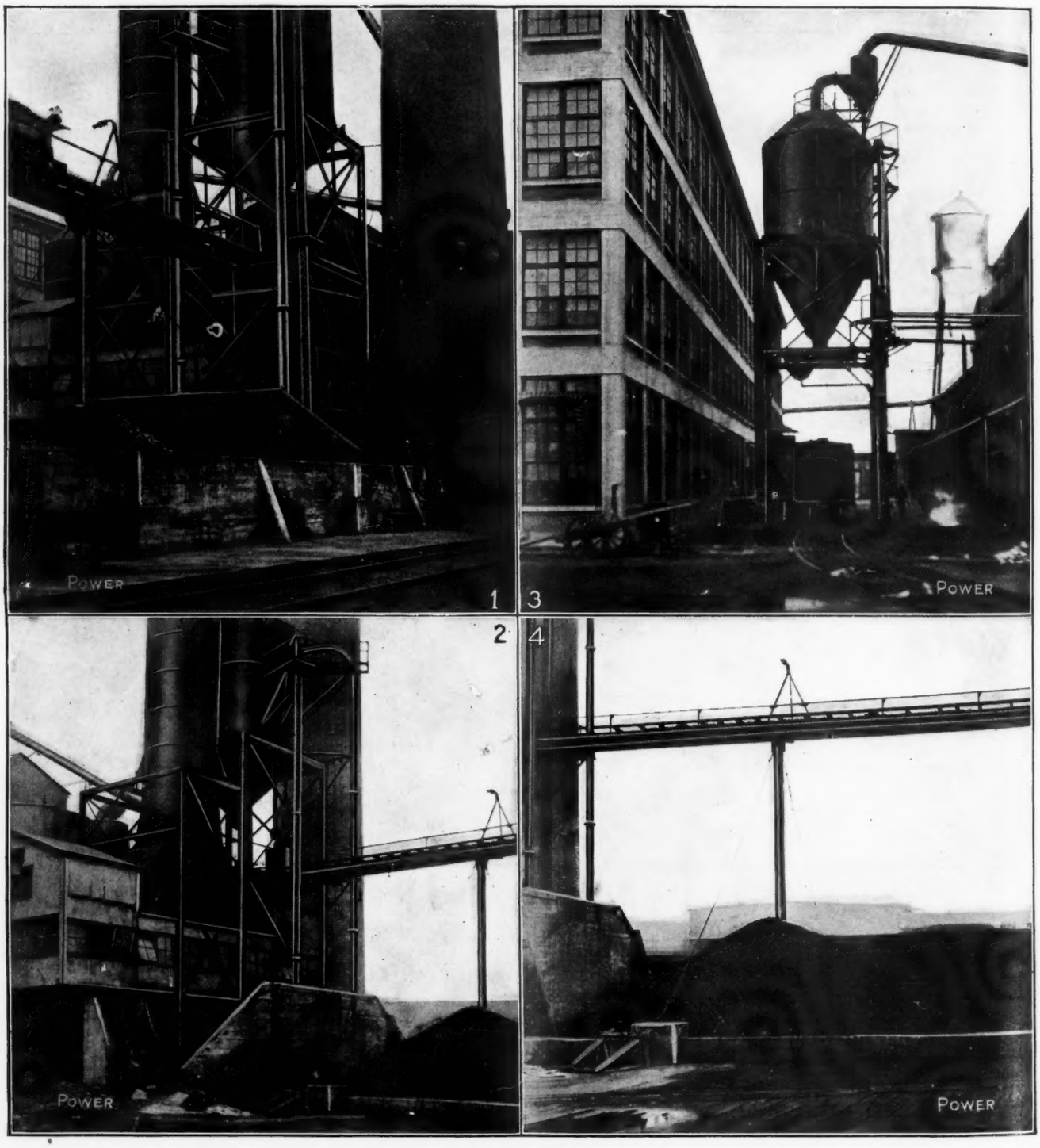
A PROSPECT

Coal and Ash Handling at Pierce-Arrow Plant

By Charles H. Bromley

SYNOPSIS—The plant uses 28,000 tons of coal a year, which averages 12 per cent. ash; all coal and ashes are handled by suction. Coal in an outside storage bin is sucked from beneath and elevated to a 150-ton tank from which it is dropped into an electric car and conveyed to the bunker, and finally discharged to the stokers. Ashes are drawn from the ashpits through 8-in. pipes to a tank of 90-ton capacity from which they are dropped directly into cars. Some operating troubles of the system are described.



Views of Coal- and Ash-Handling Equipment, Pierce-Arrow Motor Car Co.

The handling of ashes by suction is not new nor is it a rare practice, but the handling of coal and ashes by this system, and on a large scale is uncommon. At the plant of the Pierce-Arrow Motor Car Co., Buffalo, N. Y., 28,000 tons of coal averaging 12 per cent. ash is consumed annually. All coal and ashes are handled by application and modification of the Darley system to suit the conditions of location, storage, etc. At present nut coal is being used and this is handled as easily as slack.

Principal Parts of the Equipment

Essentially the system consists of a turbine-driven, (65-hp.) exhauster, which exhausts the air from large pipe lines leading from hoppers, for coal and ashes respectively, to elevated steel tanks. The coal tank, Figs. 1 and 2, is of 150-ton capacity and the ash tank, Fig. 3 of 90-ton capacity. The conveyor pipes, 10-in. for coal and 8-in. for ashes, carry about a 3-in. vacuum, while the system is operating. The ash tank is provided with a dust collector above the main tank. In Fig. 8 is a plan view of the collector showing how the dust is whirled and thrown outside the baffles to settle, while the air passes up through the center of the collector.

The coal is dropped from the tank into an electric car which carries it to the 75-ton bunkers over the boilers or to outside ground storage, Fig. 4, if the bunkers are full. From the bunker the coal is dropped directly to the stokers through the chutes.

Coal Enters Conveyor Pipe at High Velocity

Ordinarily the coal is dropped directly from the car into a feeder hopper in the conveyor pipe, but at the Pierce-Arrow plant short belt conveyors are provided to convey the coal at rather high velocity into the open end of the conveyor pipe, which is the most inefficient part of the system, owing to the inertia of the coal and eddy currents there. It is claimed that much more coal can be handled in this way than where it enters the feeder hoppers at a low velocity. Notice in Fig. 5 that there are two belt conveyors, one to take coal from the track hopper to the second belt which discharges it into the conveyor pipe. Originally these belt conveyors were not used.

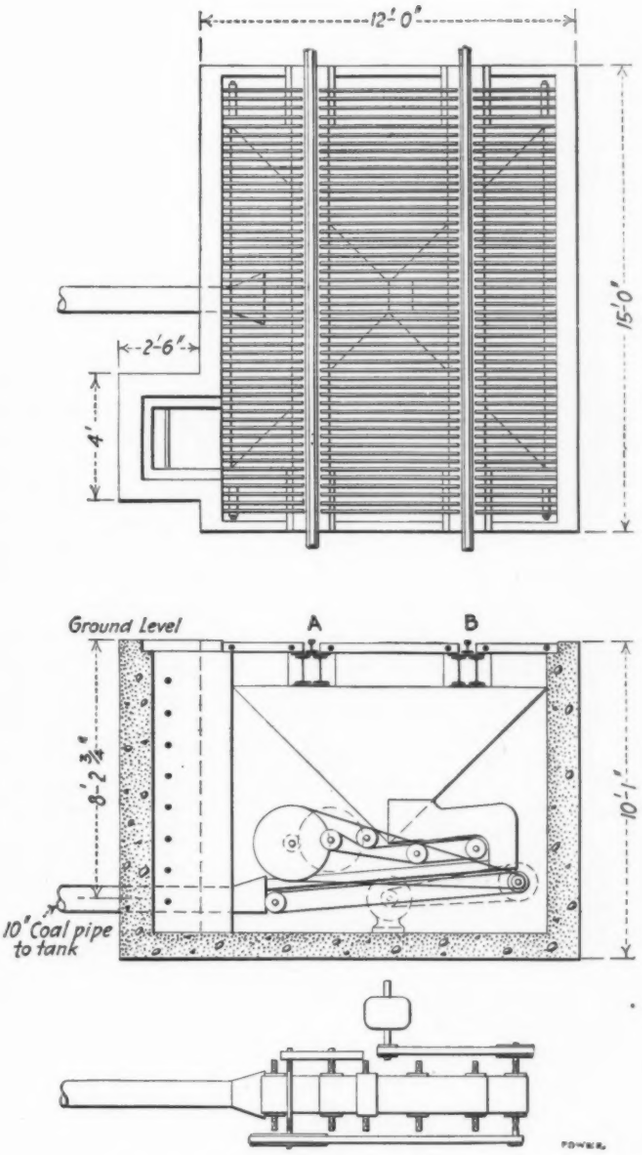


Fig. 5. Track Hopper Screen and Belt Conveyors

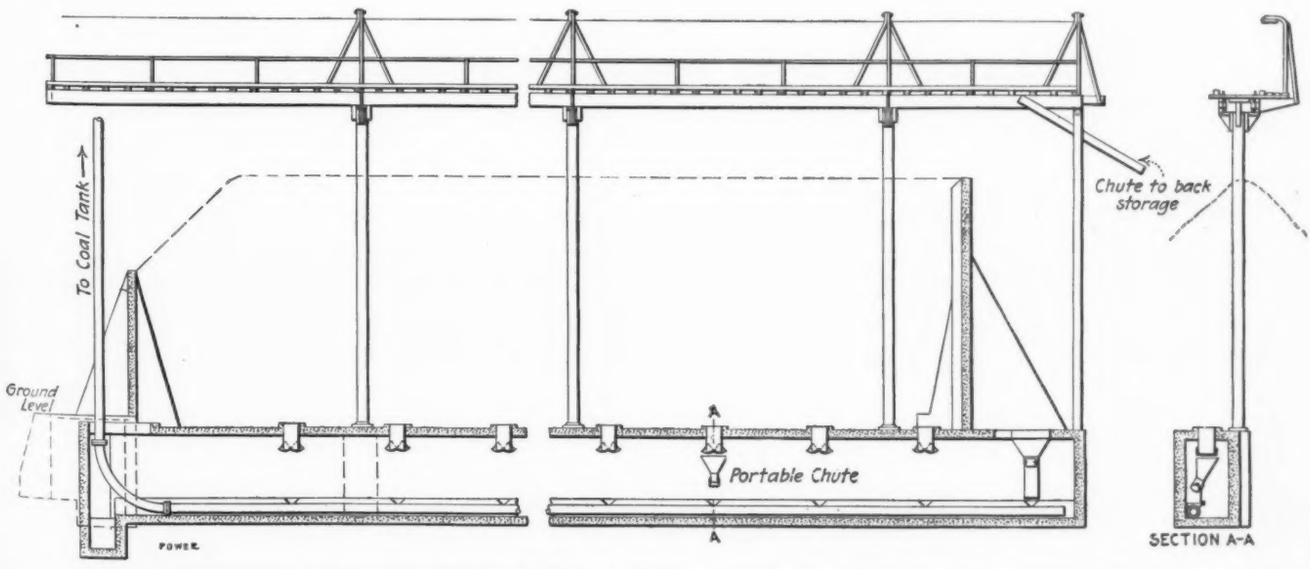


Fig. 6. Elevation of Ground Storage Bin, Showing Concrete Tunnel, Conveyor Pipe and Chute

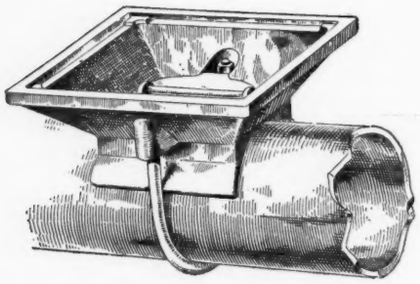


FIG. 7. DETACHABLE HOPPER

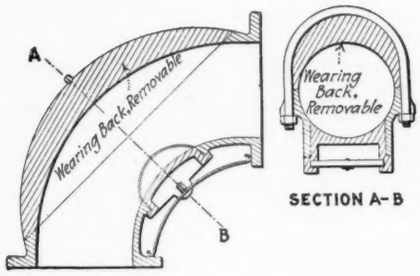


FIG. 9. REMOVABLE BACK ELBOW

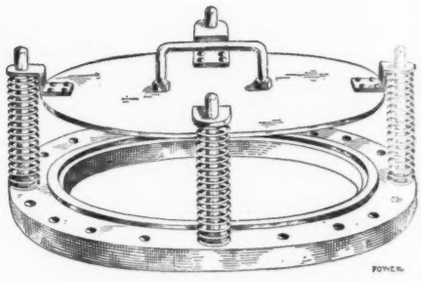


FIG. 11. PRESSURE-RELIEF COVER FOR ASH TANK

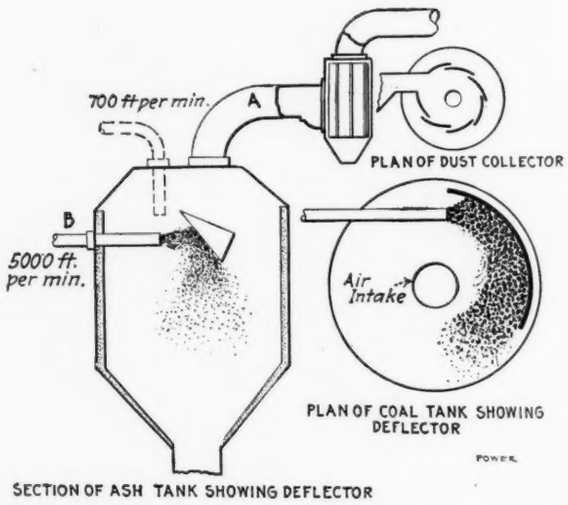


FIG. 8. DUST COLLECTOR AND ASH AND COAL DEFLECTORS

HANDLING THE STORAGE COAL

The storage part of the system and the manner of handling the coal are rather interesting. As shown in Fig. 6, beneath the ground storage area is a concrete tunnel containing a 10-in. pipe leading to the coal tank and having detachable hoppers, Fig. 7, which are nearly in a center line drawn longitudinally through the coal pile above, and below openings in the floor of the storage bin. As the opening in the bin floor is not quite in line with the feeder hoppers in the 10-in. pipe, a portable hopper chute is provided in the tunnel to guide coal from above into the hoppers from which it is conveyed by suction to the coal tank. The floor of this tunnel has a slight slope to drain off what little water may get in.

The capacity of the ground storage area connected with the 10-in. suction pipe is 2000 tons, but another storage area of 2200 tons capacity not connected with the suction system is provided for emergencies. Not long ago the coal in storage took fire and it is interesting to know that the hot coal was sucked into the tank without causing any trouble.

Some readers may wonder why the coal or ashes coming into the tank are not carried over into the air duct and then to the blower, causing damage. The following explains this:

The air duct *A*, Fig. 8, is of much larger diameter than the ash pipe *B*. By connecting the larger pipe to the blower, the air velocity in the pipe *A* will be about 700 ft. per min. and that in the pipe *B*, about 5000 ft. per min. The coal or ash is too heavy to be carried up into

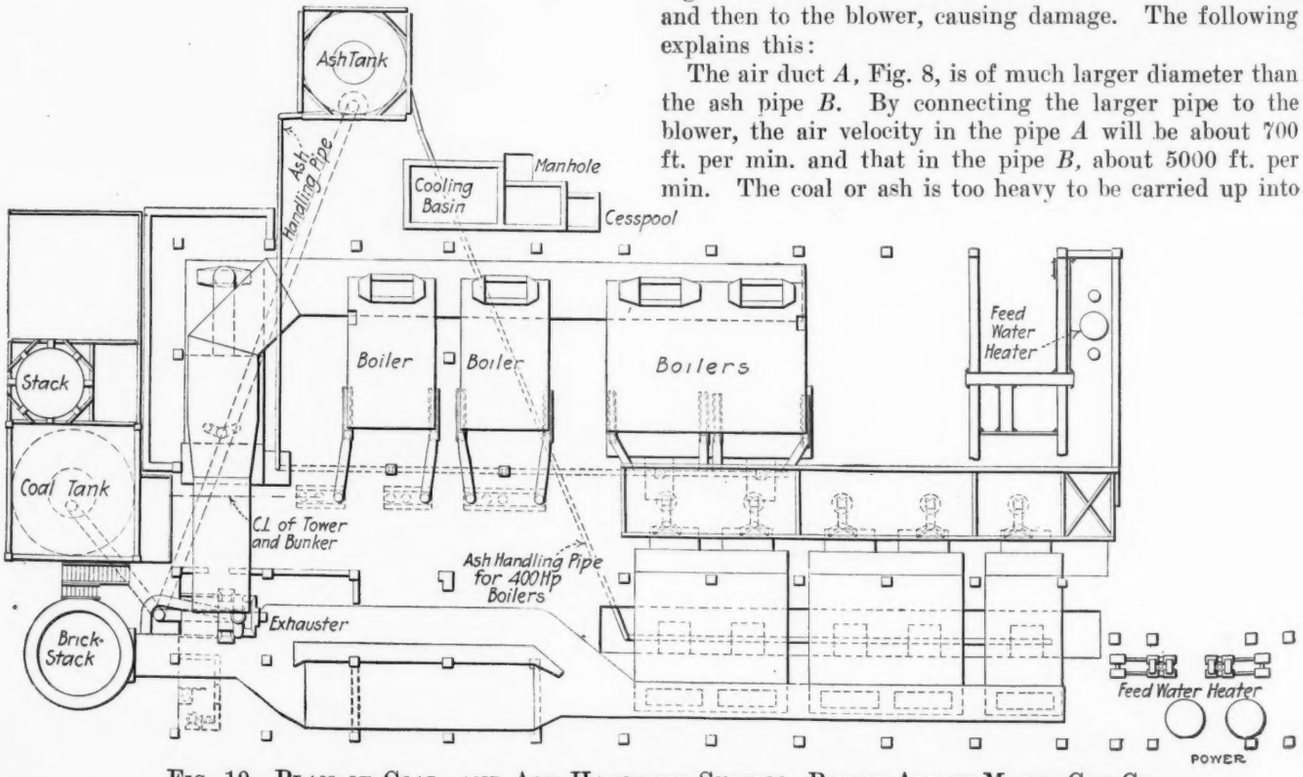


FIG. 10. PLAN OF COAL- AND ASH-HANDLING SYSTEM, PIERCE-ARROW MOTOR CAR CO.

the air duct when the air velocity is so low, even when it enters the tank tangentially as it does in this installation. Often the coal and ash pipes come into the tank from above, as shown by the dotted lines.

Provision for Wear of Conveyor Pipes

The wear on the elbows or bends in the conveyor pipes, whether for coal or ash, is considerable and so called removable wear-back elbows, Fig. 9, are used in nearly every installation. At the Pierce Arrow plant, these elbows were discarded for long sweep bends of special make so as to obtain a thickness of 2⅜ in. at the back. It was found that much more coal and ashes could be handled in a given time by the use of the bends, owing to less pipe friction. To prevent wear at the sides of the tanks, a concrete lining is often used.

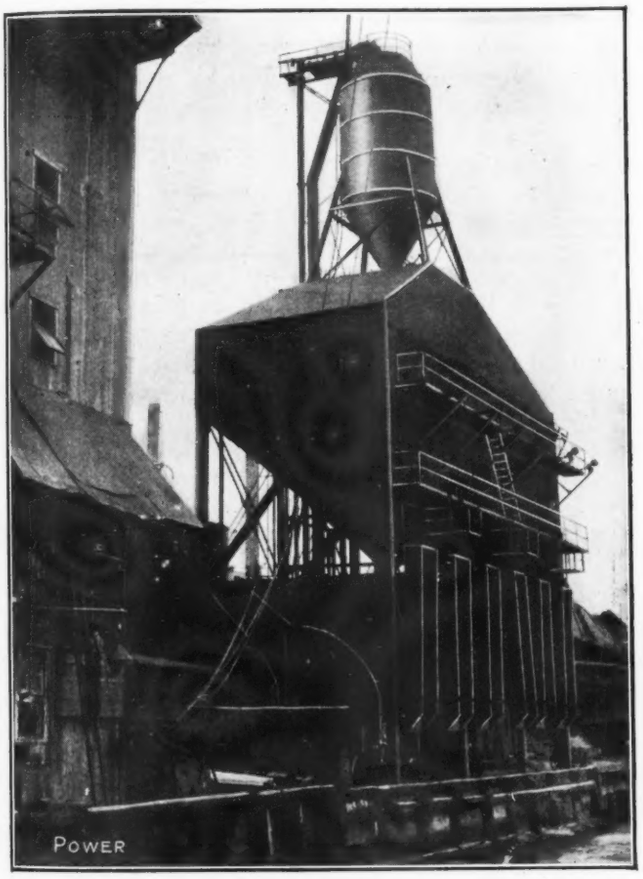


Fig. 12. Ash-Storage Bin on Water Front

There are two rows of boilers, five in each row, and originally the ash ducts under each row connected into a Y-fitting from which one pipe continued on to the tank. The Y and single pipe near it wore so rapidly that it became necessary to put in two separate pipe lines, one from each row of boilers direct to the ash tank, as shown in the plan of the system, Fig. 10, and since then no trouble has been experienced.

Explosions in Ash Tanks

When ashes are handled by this system, explosions sometimes occur, owing to an explosive mixture being formed in the tank by gases from the hot ashes mixing with air, a live coal or spark igniting the mixture. To provide against damage by such explosions, a relief cover shown in Fig. 11, is used on the top of the tank. The cover is free to be blown entirely off the guides, and springs are used to hold it off its seat when the tank is not in operation. One such explosion has occurred at the Pierce Arrow plant but no damage was done. A water line connected into the ash pipe near the tank is used to wet and cool the ashes. It is when this water line becomes plugged, frozen, drained or the valve closed that an explosion is likely to occur.

To prevent the moist ashes in the tank from freezing and binding together a ½-in. steam line is connected into the ash outlet of the tank.

At the Pierce Arrow plant, the engineers have tried to handle soot from the boilers with the system, but the water used to keep the soot moist enough to prevent it floating promiscuously throughout the system makes it form a putty-like mass that is so difficult to handle that the system is of no advantage over hand and shovel methods. In some plants soot is handled successfully, but where the soot is fine and powdery instead of granular it will plug the line after being wet. This could undoubtedly be overcome by admitting the water at, say, 2 ft. from the open end of the pipe in the tank, and by dropping the pipe vertically into the tank so that the weight of the wet soot would cause it to drop out of the pipe by its own weight.

As previously stated, the ash intake hoppers are usually at the front of the boilers, so to handle soot by the system a long heavy rubber hose with an intake nozzle like those used in vacuum cleaner systems, connects the soot chambers with the suction pipe.

Fig. 12 shows a type of storage bin used with an ash tank. This photograph was not taken at the Pierce Arrow plant but it is deemed of sufficient interest to warrant reproduction. From the bin the ashes are loaded into boats.

The system at the Pierce Arrow plant was installed by the Guarantee Construction Co., 140 Cedar St., New York City.

Fulton-Tosi Oil Engines

Perhaps the latest entrant to the growing class of builders of Diesel-type engines in America is the Fulton Iron Works, St. Louis, Mo., which is building a line of oil engines after the general design of Franco Tosi, Legnano, Italy, a builder of poppet-valve steam engines, steam turbines and gas and oil engines. A side view (Fig. 1) and a perspective (Fig. 2) of a 150-hp. four-stroke-cycle vertical unit on the testing floor of the Fulton Iron Works are given herewith; the general design is shown in Fig. 3.

Following the general European practice the engine is built with an A-type frame; all the valves, each in a separate cage, are grouped vertically in the cylinder head; the camshaft is situated on a level with the tops of the cylinders, and the air compressor is directly connected. The camshaft is driven from the main shaft by means of the vertical shaft *B*, Fig. 3, and two pairs of helical gears. The governor which controls the delivery of the fuel pump is mounted on this vertical shaft at *C*.

The fuel pump, located at *A*, Fig. 3, and shown in detail in Fig. 4, is of a special patent design developed by H. R. Setz, chief engineer for the Fulton Iron Works.

The principle of this pump is shown in Fig. 5, although the construction is not identically the same as that in Fig. 4. Oil enters the pump through the suction valve *D*, Fig. 5, and is forced out by the plunger *E*, through the discharge valve *F*. As the stroke of the plunger is always constant its displacement must be likewise. Consequently, the quantity of oil pumped at each stroke is varied by

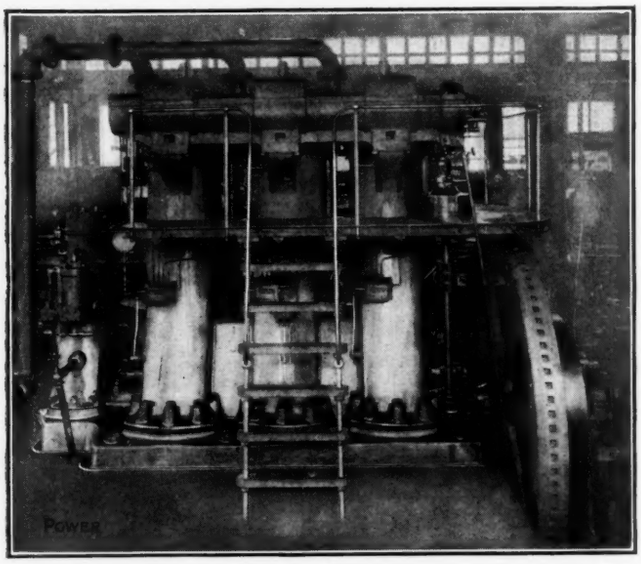


FIG. 1. 150-HP., THREE-CYLINDER, FOUR-STROKE-CYCLE FULTON-TOSI OIL ENGINE ON TESTING FLOOR

varying the clearance volume of the pump chamber *G*. This is done by means of the vanadium-steel diaphragm *H* behind which are two spring-steel disks for reinforcing. The play of the diaphragm is limited by the plunger *I* whose position is controlled by the wedge *J* connected with the governor *K* in the manner shown. When the engine speeds up the governor rises, thereby partly withdrawing the wedge *J* and allowing the diaphragm greater play. The in-stroke of the pump plunger, then, instead of forcing the full measure of oil out at the discharge valve against the high back pressure encountered, forces out only the required fraction, while the bulging of the diaphragm takes care of the balance of the displacement. On the out-stroke the diaphragm returns to the normal position, thereby preventing admission of fuel in excess of that required. In the pump as actually constructed there are two diaphragms instead of one and these are at the sides of, instead of facing, the pump plunger.

The fuel-injection valve of each cylinder is placed in a vertical position and in the center of the cylinder head, thereby making it possible to concentrate the bulk of air necessary to sustain combustion uniformly around the injection nozzle. A perfectly symmetrical and cylindrical combustion chamber is thus obtained which is conducive to maximum efficiency and insures easy control of the stresses to which the cylinder head is subjected. The mechanical clearance between the piston top in its highest position and the cylinder head is over 1 in., thus positively preventing contact between the piston and the cylinder head.

The principal parts of the fuel-injection valve are the atomizer, which consists of a series of perforated disks, mounted on the needle-valve guide, a grooved cone below these disks and the injection nozzle. The fuel oil is deposited on these disks by the pump while the needle valve is closed. This gives the oil time to distribute itself in a thin uniform layer over the whole surface of the disks so that when the needle valve opens the oil is swept from this large surface into the cylinder by the injection air.

The air and exhaust valves, which open inwardly, are each actuated by a single lever, the outer end of which rides on top of its cam. The fuel-injection valve, however, opens outwardly, and, to bring the lever contact on top of the cam in this case, as well as in the other two, a second small lever is employed. This has as its fulcrum a pin in the bonnet of the valve itself set eccentrically and fitted with a handle and latch, so that the small lever may be shifted to change the lead of the valve and the duration of opening at will.

Accessibility is another feature of the valve and, for that matter, of the whole engine. By taking off the spring case at the top, the fuel needle may be withdrawn without disturbing the valve setting and the atomizer parts are made accessible simply by removing two nuts.

Two of the three cylinders are fitted with valves for starting on compressed air and these, themselves, are operated by air, instead of mechanically. This arrangement offers the advantage of doing away with one cam and lever, which is important because of the increased accessibility it makes possible. With only three cam-and-lever sets to the cylinder, the levers for the admission and

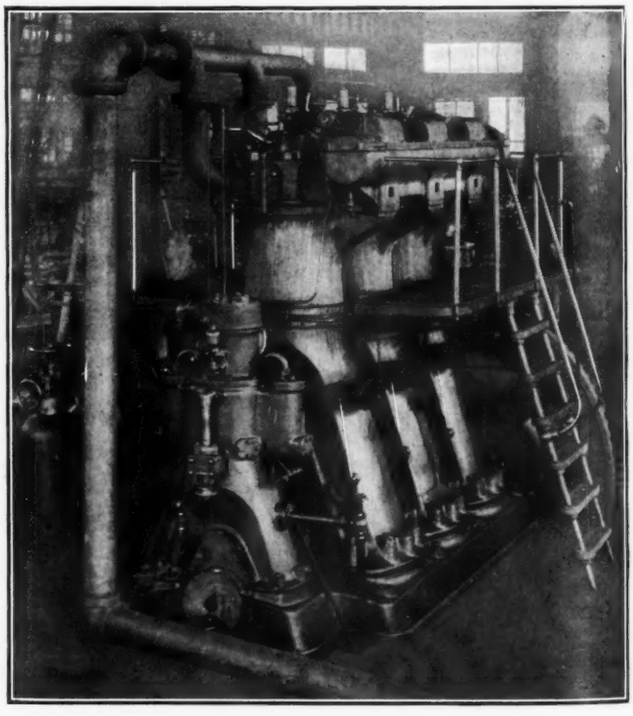


FIG. 2. ANOTHER VIEW OF THE FULTON-TOSI ENGINE

exhaust valves can be pivoted on the ends of a pin, while the larger of the levers for the fuel valve is mounted at the middle, the pin being supported at either side of the middle lever. Thus, the exhaust- or admission-valve lever can be removed simply by taking out a tap bolt from the end of the pin.

With air-operated starting valves the control is concentrated at one point so that the process of starting is reduced to a simple operation. By means of a rotary distributor, which is thrown out of operation after the

engine is started, high-pressure air is admitted to the top of a piston mounted on the stem of the balanced starting valve. This raises the starting valve and permits the starting air to enter the cylinder during about 45 per cent. of the downward stroke of the piston, then the closed spaces of the crank cases so that the air drawn in may be somewhat higher in temperature than that outside and the crank and rods may enjoy the benefit of resulting ventilation. On the smaller sizes of engines all the main bearings are lubricated by means of oil rings,

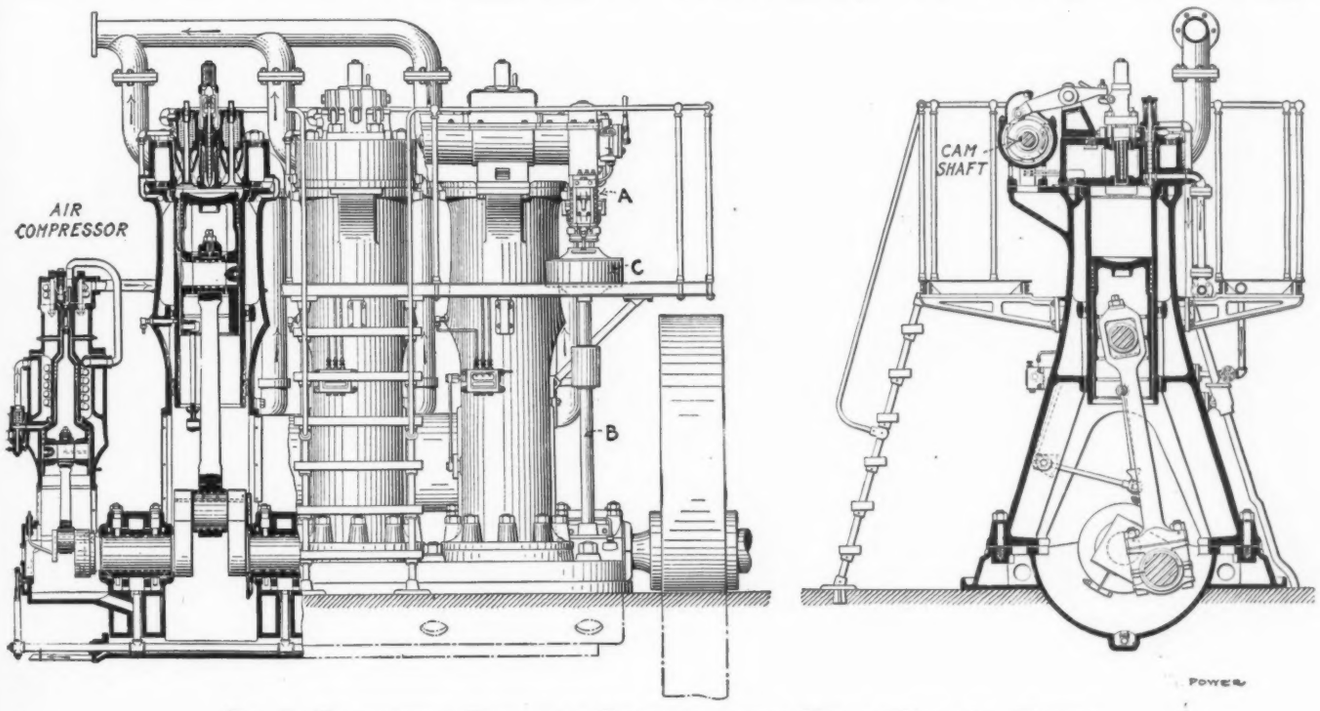


FIG. 3. ELEVATIONS, PARTLY IN SECTION, OF THE FULTON-TOSI OIL ENGINE

rotary distributor establishes a connection from the top of the starting valve to the atmosphere, which brings the valve back to its seat.

The cylinder design takes the form of a heavy liner, the space between which and the outer casting comprises the water jacket. The admission valves are piped to the in-

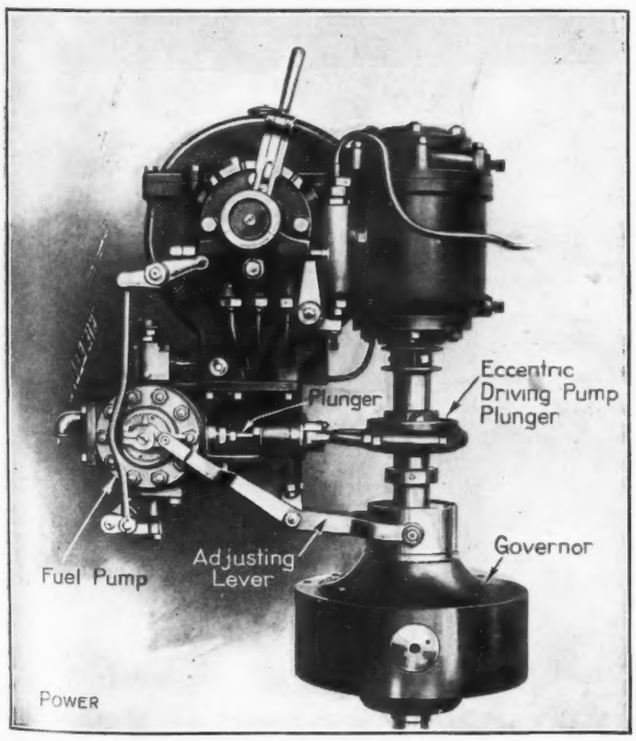


FIG. 4. GOVERNOR AND FUEL PUMP OF THE ENGINE

while the crankpins receive oil from a centrifugal ring, as customary on gas engines. On the larger engines the main bearings as well as the crank- and piston pins are lubricated by a rotary pressure pump. A separate sight-feed oil pump is provided for each piston which forces

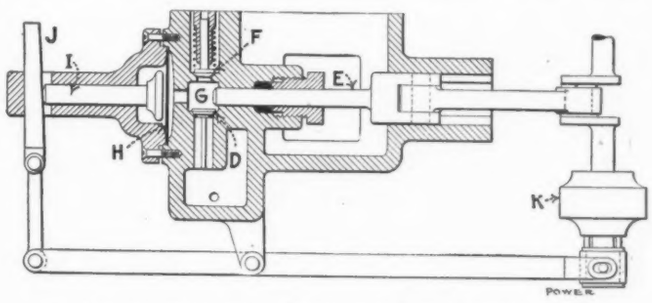


FIG. 5. SHOWING PRINCIPLE OF FUEL PUMP

the oil into the cylinder at from four to eight points as the piston approaches the lower dead center.

Conforming with the best European practice the three-stage injection air compressor forms an integral part of the engine. A cooler, arranged after each stage, is built into the water space of the compressor frame, thus making it a compact self-contained unit. The quantity of the air delivered by the compressor can be varied somewhat by throttling at the low-pressure inlet, thus regulating the rate of supply to satisfy the change in demand with variation in load.

The Fulton Iron Works is now building two-, three and four-cylinder, four-stroke-cycle engines in capacities of from 100 to 800 b.hp. and will shortly be prepared to build two-stroke-cycle units of capacities ranging from 800 to 3000 b.hp.

T. Hunter, Hoisting Engineer

By Warren O. Rogers

SYNOPSIS—Hunter airs his opinions as to why mine superintendents lack ability to economically operate steam boilers. He points out why the low cost of coal, ignorance and neglect are responsible for the uneconomical conditions existing at so many mine boiler plants.

During my visits to the mines with Hunter I had noticed that most of the boiler plants were, what would be termed, if found in an ordinary manufacturing plant, in a dilapidated condition. Many of them were old and patched; the iron fronts were cracked and warped with the heat and the settings were anything but samples of economy. Of course, there were exceptions, but on the whole the boiler plants were in bad shape. No attempt had been made to keep the boiler room tidy or the coal off the floor in front of the boilers, Figs. 1 and 2.

Just what the boiler performance was could not be ascertained, because no record was kept of the evaporation of water per pound of coal, nor of anything else that I could find.

As one might expect, this condition did not exist altogether in the plants of the smaller coal companies. In many instances the engine room was some distance from the boiler plant, necessitating a long line of steam piping. In no instance could anyone give even an approximate idea of the amount of condensation in the pipe lines, generally stating that it did not amount to much.

Leaking pipe joints seemed to be the order of the day, Fig. 3, which on account of the length of the pipe lines and manner of suspension were doubtless difficult to prevent, but judging from the amount of steam escaping it must have cost a pretty penny in the course of a year's

Fig. 1. Typical Return-Tubular Colliery Boiler Plant

run. Most boiler houses contained water-tube boilers of some type.

"Hunter," said I, when after a tedious climb up a steep hill, we sat down to rest, where we could watch the loaded cars being hauled out of a slope, Fig. 4, "why is it that most mine boiler plants have the appearance of being neglected? You've had enough experience to know."

"Yes, I know. It is due to three of the commonest causes found about a coal mine; cheap coal, ignorance and neglect. One cause is perhaps as responsible as the other."

"I don't understand why any cause should be tolerated. Coal is cheap, of course, and what isn't burned could be turned in for the market, so there is no reason for wasting coal. If only the lowest grade of coal were burned, coal that could not be marketed, even that would not justify a waste of fuel, because more men would be required in the boiler room to attend to the fires."

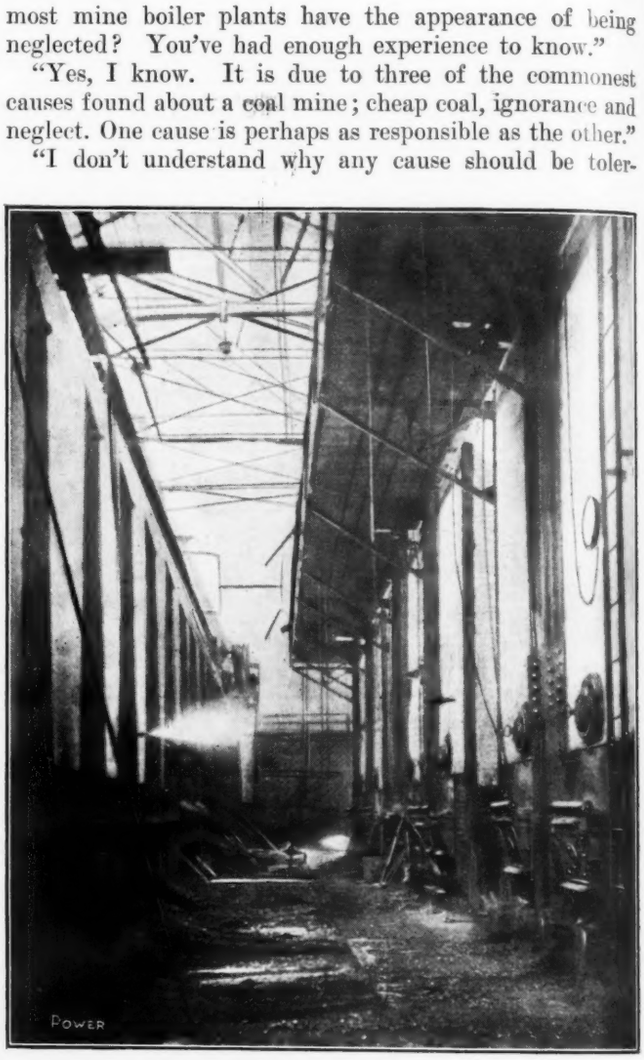


Fig. 2. An Ill-Kept Boiler Room

"Well, cheap coal is, in my opinion, the principal reason why the boiler plant is neglected. If it were necessary to pay, say, $4 a ton, you wouldn't see boilers with tubes plugged and scaled and demanding 20 per cent. more fuel than they should."

"I've seen boilers run with the greater part of the grates covered solid with clinker, because the firemen cleaned the front part of the grate only, when the fires were supposed to have been thoroughly cleaned."

"No intelligence is shown in carrying fires between 20 and 24 in. thick, slicing fires every half hour and firing twice as much coal at a charge than is necessary, believe me. We have found some of these conditions existing in the boiler houses we have visited, and in some of them all were in evidence."

"Well, if the owners don't keep things fixed up, you

can't expect the firemen to lose any sleep over the way the boilers are run. Why doesn't the superintendent get after things, he is the man in charge, I believe."

"Now you are getting at the second cause of uneconomical operation. Although the working of the mine depends entirely upon the boiler plant, the majority of mine superintendents pay but little attention to the boilers, so long as they generate enough steam. Uptodate, economical boiler operation cuts a small figure. Give the machines enough steam and the boiler plant is forgotten until something happens."

Hunter stopped talking long enough to throw a piece of slate at a rattlesnake that was sunning itself on a log some distance away, and then resumed his conversation.

FIG. 3. LEAKING PIPE JOINTS SEEM TO BE THE ORDER OF THE DAY

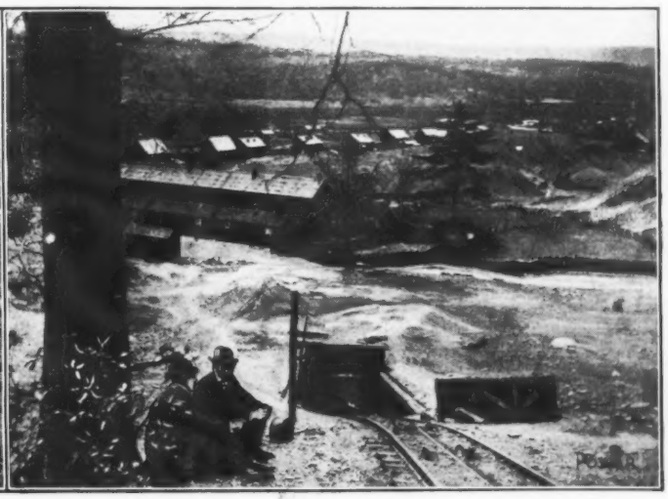


FIG. 4. WE REST AWHILE AFTER A CLIMB UP A STEEP HILL

"In every boiler room the load on the boilers, draft, condition of the fires and boilers, also the grade of fuel used, are variable factors and, unless studied, it is impossible to get good boiler performance and the fuel consumption will surely be excessive. Just let the string of cars going to a breaker stop and the superintendent is right on the job to find out what caused the stoppage of coal movement and the visible loss of output. In the boiler room, however, coal is moving. Not to the market, but into the boiler furnace. This is a different kind of stoppage, and one the average mine superintendent does not notice."

"You paint the situation rather black," I replied, "all mine boiler plants are not operated under such conditions. We have seen them in which good equipment was found and fairly good operating conditions were maintained."

"Sure, I know that, for there are superintendents who know that any kind of a boiler will not do for colliery work. We have seen a number of boiler plants, and there are plenty more, where high-pressure, water-tube boilers are used. The superintendents operating such boilers are uptodate, and recognize that a good boiler, economically operated, is a mighty good investment."

"It seems to me that if the lower-grade fuel is to be burned, the boilers would have to be designed with plenty of heating surface per horsepower rating. Then shaking grates would come in handy or even stokers."

"Right you are," replied Hunter. "The boilers of the average mine plant are equipped with shaking grates of some type. Stokers, however, are found mostly at mines having a central-power plant from which electrical energy is supplied to several tipples and shafts."

"Why are not stokers used at the average mine boiler plant?" I asked, somewhat surprised now that the matter had been brought to my attention.

Hunter, leisurely cut the end of a fresh cigar, and after lighting it to his satisfaction, and leaning back against the tree trunk with his hands behind his head, said, "What would be the use? Just as much labor would be required and practically as good results can be obtained from hand firing, if attention is given to the method employed, and if the boiler setting is kept in repair and the boilers kept clean.

"I favor shaking grates, because it is easier to burn the refuse of the mine on them, than with stationary grates. When using stationary grates the same precautions should be observed as in any boiler plant, that is, to keep the air spaces open and the grates properly fitted in place. That means the removal of warped grates, because they will allow fuel to drop through into the ashpit. Shaking grates allow cleaning the fires of ashes and fine clinkers without opening the furnace door, and prevent cold air entering the furnace to cool the gases during the operation.

"That is where the fireman comes in," said I, jumping to escape a lump of coal that had fallen from a loaded coal car, and came rolling down the hill. The superintendent can't stay in the boiler house all the time, and so it's up to the foreman to produce the goods after proper apparatus has been given him with which to work."

"Yes, it is up to the fireman, and superintendent also. If the superintendent does not know enough to put intelligent firemen in front of the boilers, he can't expect to get good results, even with first-class apparatus. You might give a small boy a dollar watch to play with, but you wouldn't even think of letting him so much as touch a $50 one. You know about what would happen if you did.

"Now, putting an ignorant fireman in **front** of a boiler is as foolish from a practical and economical standpoint as giving a boy a $50 watch with which to play. Both will make a mess of it. Here is where the low cost of fuel comes in. The superintendent says, 'What is the use of

paying a fireman high wages when the coal he burns costs so little? He loses sight of the fact that the coal burned in the boiler furnace is worth just as much a ton as the coal that is loaded into cars and shipped to the market.'

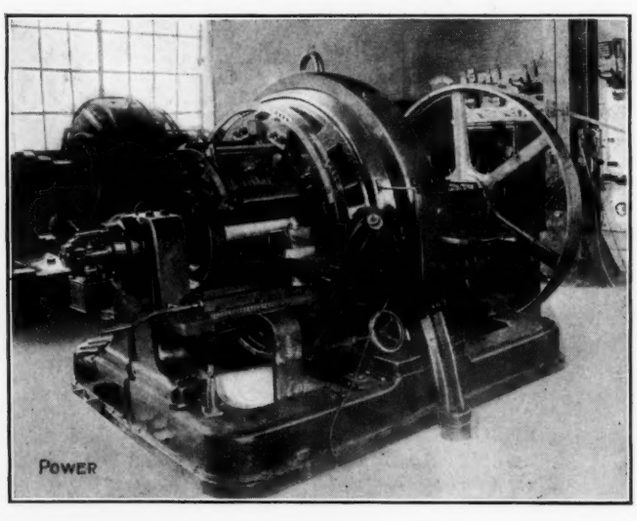
"Yes, I have heard it said that a good fireman can save his pay in the saving of coal over what an inefficient fireman would burn. If that is so, why is not the fireman about the most important man about the plant, although giving the engineers all due credit?"

"He is in many respects," replied Hunter, arising and stretching himself. "Come on, let's hike for the hotel and supper. I think I've earned a good one today. At the table, Hunter's actions did not belie his words, and to tell the truth, I was a close second.

Rig for Truing Commutators

By A. D. Williams

The commutator-truing rig herewith shown is used in the West Breakwater plant of the Pennsylvania R.R., at Cleveland, Ohio. It was designed by J. J. Weick, foreman of the plant, and possesses some novel features, enabling one man to do the work after the rig is in place. A large belt wheel is fixed on the projecting end of the shaft at the slip-ring side of the machine and belted to a



Truing Rig in Position

5-hp. direct-current motor. This motor is mounted on a truck with a speed-controlling rheostat and a magnet-switch starting and stopping control, this last being operated from the small control switch on the flexible cord alongside the five-pole double-throw switch on the field frame of the rotary converter. This enables the man working with the rig to stop and start the motor without trouble and the leads to the control switch are long enough to permit his carrying it anywhere around the machine. The large belt wheel is bored to fit the shaft of a large flywheel generator set, which can be seen just beyond the rotary converter, and is bushed down to fit the smaller shafts of the converter and the exciter sets.

At the time the photograph was taken the machine had just been in use smoothing a commutator with a sandpaper block, which is shown in place of the tool holder. The frame for the rig is made of steel bars, about ¾x6 in., and the tool holder is similar to that used on a lathe having a compound hand feed for regulating the depth of cut, and in addition the tool holder can be fed along the ways by the crank shown just below the oscillating device of the converter. The ways are supported at three points, by a bracket fitting onto the brush-holder ring of the machine, by a steel bar strut reaching down with a jack-screw and block resting on the floor, and by the Z-bar support to the top of the pillow block. This Z-bar can be made of different lengths to suit other machines. Slotted holes can be used where necessary to permit the adjusting of the tool guide parallel with the shaft of the machine upon which it is to be used, but in this plant the rig has been made with parts to fit the different machines on which it will be used, so that it can be quickly adjusted to them. The rig requires only the removal of two sets of brush-holders in order to set it up.

Zigler-Holden Safety Switch

The accompanying figures illustrate the Zigler-Holden safety switch which may be used for either alternating- or direct-current service and can be supplied in almost any size desired. The switch is furnished in two types, with or without fuse compartment and clips. Fig. 1 shows a switch installed and Fig. 2 shows the design of a switch with fuse compartment. Where fuses are included the cover is divided into two parts so that the fuse-compartment cover may be removed without exposing the switch itself.

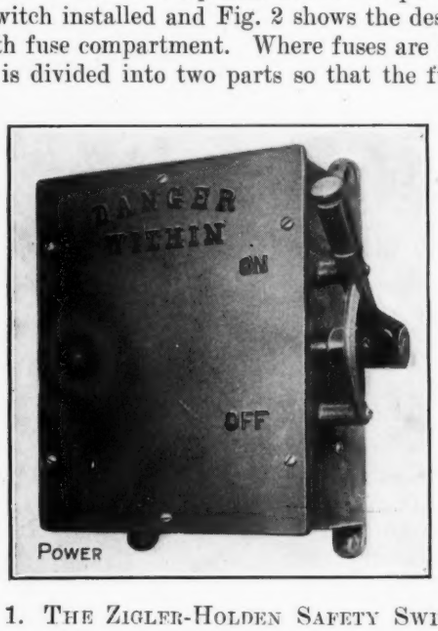


Fig. 1. The Zigler-Holden Safety Switch

The switch blades are mounted on either a hard-wood or porcelain roller journaled in the sides of the box which is of heavy cast iron. The switch is thrown by means of the outside handle *A*, Fig. 2, fitted with the latch *B*. To close the switch the latch must be released and the handle raised the full distance and latched in place. If the handle is not latched in the closed position a recoil spring attached to the roller and the weight of the handle will cause the switch to open and automatically latch itself in the open position. This arrangement makes it impossible for the switch to remain in any but the fully open or closed positions unless it is held. The tension of the spring can be easily adjusted without opening the switch box.

This switch seems as nearly fool proof as it is possible

to make such a device, and should be of great value in plants where electric drive makes it necessary that switches be often manipulated by men who are more or less ignorant of the dangers of electric circuits. The design makes it practically impossible for the contacts to burn or fuse and the absence of hinges and hinge belts in the path of the current makes for perfect electrical connection. Its fireproof construction is another point to commend it. Being all standardized the switch parts may easily be renewed.

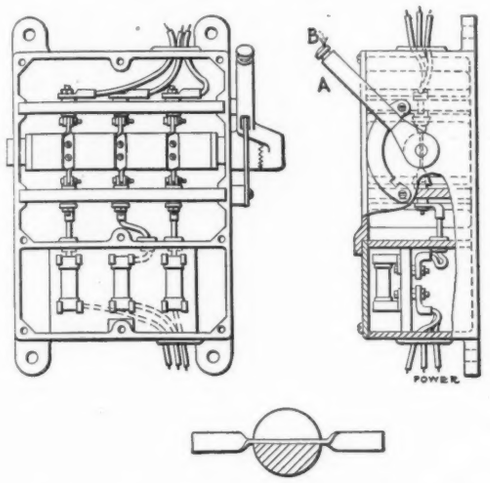


FIG. 2. DETAILS OF THE INTERIOR OF THE ZIGLER-HOLDEN SWITCH

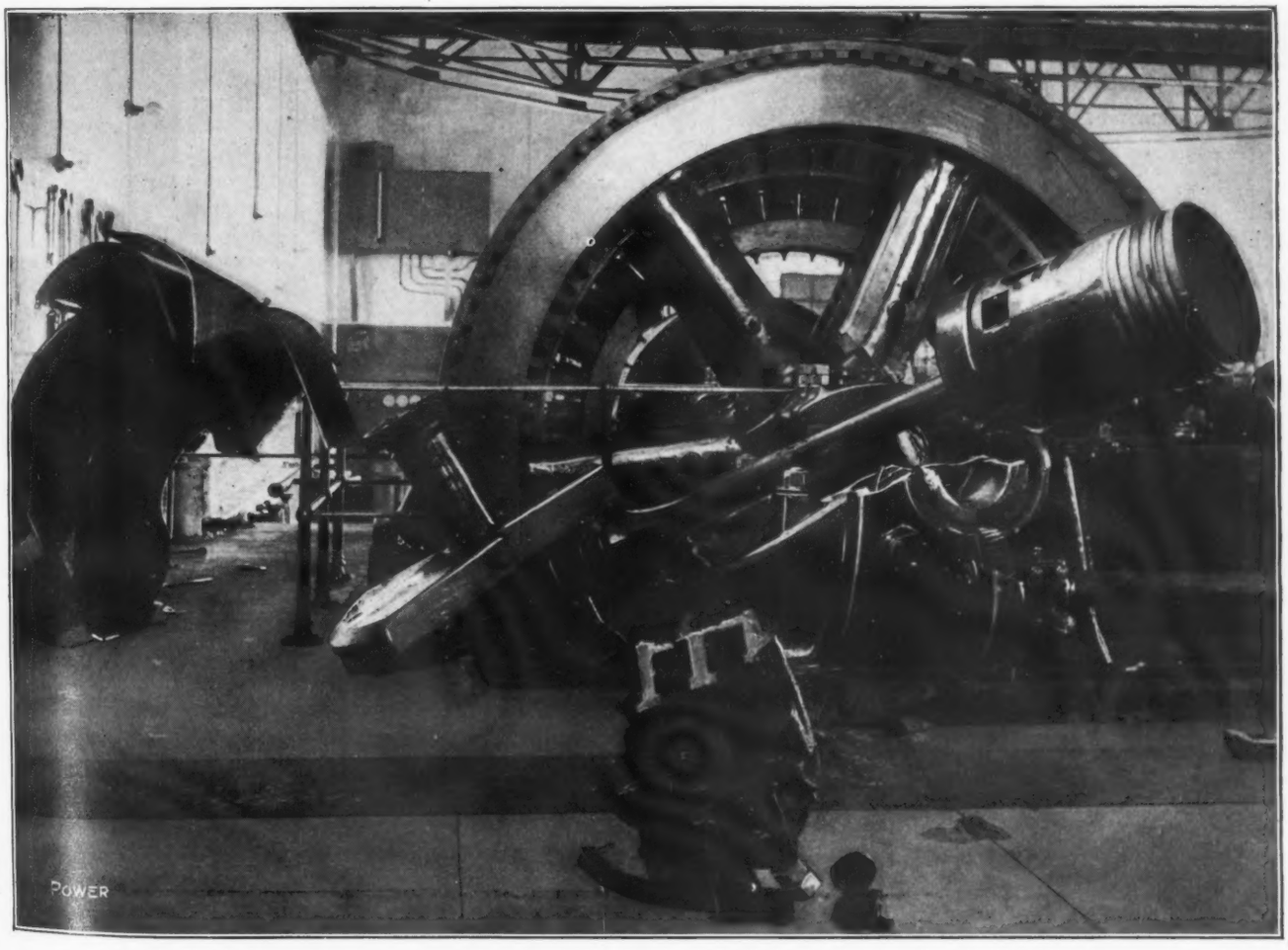
The patents on, and the manufacture of, the Zigler-Holden switch are controlled by the Automatic Safety Electric Switch Co., 1037 Waller Ave., Chicago, Ill.

Wrecked Gas Engine in Mexican Power Plant

BY W. A. WAYLAND

The photograph shows the damage resulting from failure of the main shaft of a 250-hp. Koerting gas engine at the light and power plant at Aguascalientes, Mex. The engine which is of the twin type, running at 180 r.p.m., had been in operation for about five years, giving satisfactory service, and on the night of the accident was in charge of a native mechanic. Although both the crankshaft and the upper half of the bearing were broken off and the piston pulled out of the cylinder, as shown, the wreckage fell clear of the flywheel and the engine continued to run on the other cylinder till morning.

The initial failure appears to have been due to an old crack extending through almost the entire section of the inner crank, the engine apparently having been at work for some time on about 7/8 in. of metal. Nobody was injured.



WRECKED ENGINE, SHOWING PISTON PULLED OUT OF CYLINDER

Measuring the Stresses in a Boiler Shell

The measurement of stress was the subject chosen by Prof. E. G. Coker for the first Kantor lecture of the present session of the (English) Society of Art. The opening lecture was devoted to mechanical methods of measurements, and included an account of the highly ingenious and instructive experiments on rubber models carried out by J. S. Wilson and W. Gore, and Professor Pearson's experiment made with models of jelly. He showed how the shortening of the columns, with the addition of successive stories to the Bankers Trust Co.'s Building in New York, was measured by an extensometer capable of detecting changes of length of 0.0002 in. The portion of his lecture of most immediate interest to POWER readers was his account of a remarkable series of strain measurements made on the steam drum of a Babcock & Wilcox boiler, by S. H. Barraclough and A. J. Gibson, described in a paper printed in the proceedings of the Engineering Association of New South Wales, sessions of 1910 and 1911. Mirror extensometers were used, enabling very accurate measurements to be made even with a very short distance between the gage points. A second mirror was fitted to record by its tilt any flexure of the part under observation. The results are represented in the accompanying diagram reproduced from *Engineering*. It will be seen that the stress is far from equably distributed in the neighborhood of the lap joints, and that it rises to very high values at the corners of the bumped heads where the radius changes for the flange.

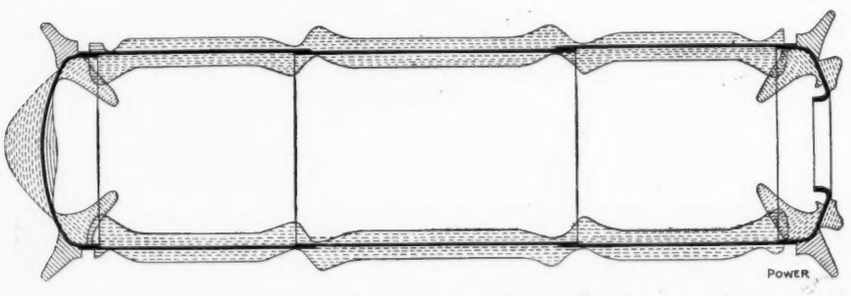


DIAGRAM OF STRESSES IN A BABCOCK & WILCOX BOILER DRUM

Compression or Absorption Ice Plant

Before the Southwestern Ice Manufacturers Association, at the annual convention in Marlin, Tex., L. C. Nordmeyer read an interesting paper on the above subject. The author realized that in a short paper it was impossible to compare these two types of plant for any and all conditions. A selection was made of the conditions and comparative data given of two 50-ton plants. As most of the refrigerating plants in the Southwest are ice plants, and as the great majority operate during the summer months with high-temperature water, an ice-making plant using a cooling tower delivering water at an average temperature of 85 deg. F. for condensing and cooling purposes was chosen. The capacity of 50 tons was selected, as this seems to be the average size of plant in this vicinity. The conditions on which the investigation was made are as follows:

Heat value of coal, B.t.u.	12,000
Efficiency of boiler, per cent.	60
Temperature of condensing water from tower, deg. F.	85
Source of supply	City Water
Water circulated per minute for condensing purposes in compression plant at 4.5 gal. per min. per ton	225
Water circulated per minute for condensing purposes in absorption plant at 9.5 gal. per min. per ton	475
Height to which water must be pumped in each case, ft.	60
Hp. required for pumping water in compression plant	6
Hp. required for pumping water in absorption plant	11
Cooling tower capacity for compression plant, gal.	324,000
Cooling tower capacity for absorption plant, gal.	684,000
Water loss through evaporation in each case, per cent	10
Hp. to drive cooling tower fan, compression plant	5
Hp. to drive cooling tower fan, absorption plant	12
Hp. to drive freezing tank agitator	4
Hp. for electric light and air compressor	10
I.hp. of aqua pump	9
Cost coal per ton (assumed in bins)	$2
Cost 1000 gal. city water (assumed) in cents	10
Cost cooling tower in compression plant with motor	$2660
Cost cooling tower in absorption plant with motor	$4450
Freight rate (approximate per 100 lb.) in cents	97
Salary to day engineer per day	$3.33
Salary to night engineer per night	$2.50
Wages paid ice pullers per day	$1.75
Wages paid firemen per day	$2.00
Interest on money invested, per cent	6
Depreciation and obsolescence, per cent	5
Insurance and taxes, per cent	1½

Of the average ice-making plant, the annual output is approximately 60 per cent. of the total possible annual capacity, so that in establishing cost per ton of ice produced, as affected by charges which are not directly as the tonnage, 60 per cent. of the possible output has been taken as the quantity of ice made. Thus:

$$365 \times 50 \times 0.6 = 10{,}950 \text{ } \textit{tons per year}$$

The cost of the building, including boiler and engine rooms, freezing tank room and daily ice storage, complete, including insulation, will be approximately $20,000. For either type of plant the fixed charges on the building will be: Interest, 6 per cent.; insurance and taxes, 1½ per cent.; depreciation, 5 per cent.; making a total of 12½ per cent. on $20,000, which would be $2500 per year. As the total ice output for the year is 10,950 tons, the building charge per ton will be

$$\$2500 \div 10{,}950 = \$0.228$$

Compression Plant

In the compression plant the mechanical equipment includes a standard compressor and complete apparatus for freezing distilled water in the cans, including all insulation brine and ammonia, three return-tubular boilers one being a spare unit, together with required brickwork and stack, water-circulating pumps, a 300,000-gal. cooling tower, a 20-kw. engine and generator. All the auxiliary machinery except the boiler-feed pump is to be electrically driven. The water-circulating pumps are to be in duplicate, and one of these is to be steam driven. The cost for the mechanical equipment erected, including foundations and engineering charges, complete, will be $42,265. The total cost of the plant, including building and machinery, will thus be $62,265. There will be burned 11½ tons or 23,000 lb. of coal in 24 hr. when producing full capacity, so that the fuel per ton of ice will cost

$$\frac{23{,}000 \times 2}{2000 \times 50} = \$0.46$$

The operating cost was estimated as follows:

Two firemen at $720 per year	$1,440
Two engineers	2,100
Handy man	720
Oil, waste, small supplies, etc. (includes 25 per cent ammonia loss)	450
Total	$4,710

Dividing the $4710 by 10,950 gives a cost per ton of ice of $0.43. Adding to this 7c. for ice handling makes a total operating expense, exclusive of coal, of 50c. per ton of ice.

In obtaining the fixed charges on the mechanical equipment, an allowance of 6 per cent. was made for interest on the machinery investment, 1½ per cent. for insurance and taxes, and 5 per cent. for depreciation and obsolescence, making a total of 12½ per cent. on $42,265, or $5283. Dividing by the annual output of the plant gives $0.482 per ton of ice as the fixed mechanical charge.

In the cooling tower, the water loss which must be made up, will be 10 per cent. of 324,000 gal., or 32,400 gal. per day. At 10c. per 1000 gal. this will cost $3.20 per day, or $0.064 per ton of ice. Collecting the above items gives the following total cost of ice per ton:

Water	$0.064
Fixed mechanical cost	0.482
Fuel	0.46
Operating expense	0.50
Total	$1.506
Fixed building cost	0.228
Total cost	$1.734

Absorption Plant

In the absorption plant the same average annual output of ice as compared with the capacity of the plant, is considered, and the same fixed building charge of $0.228 per ton of ice. The mechanical equipment of the plant includes a complete standard absorption machine and apparatus for freezing distilled water in cans, using evaporators to supply the necessary distilled water not supplied by other apparatus, all insulation, brine and ammonia included, two return-tubular boilers, one being a spare, together with required brickwork and stack, water-circulating pumps, a 700,000-gal. water-cooling tower, and a 30-kw. engine and generator. All the auxiliary machinery except the boiler feed and aqua pumps is to be electrically driven, and as before, the water-circulating pumps are to be in duplicate, one of them being steam driven.

The cost of the mechanical equipment erected, including foundations and engineering charges, will be $46,845, and the total cost of the plant, building and machinery will be $66,845. To produce full capacity will require 15,480 lb. of coal in 24 hr., making the fuel cost per ton of ice

$$\frac{15{,}480 \times 2}{2000 \times 50} = \$0.3096$$

The operating cost is given in the following:

Firemen	$1,440
Engineers	2,100
Handy man	720
Oil, waste, small supplies and ammonia	300
Total	$4,560

The total, $4560, divided by the annual output of ice, gives a charge of $0.416 per ton of ice. Adding 7c. for ice handling, makes the operating expense $0.486 per ton of ice.

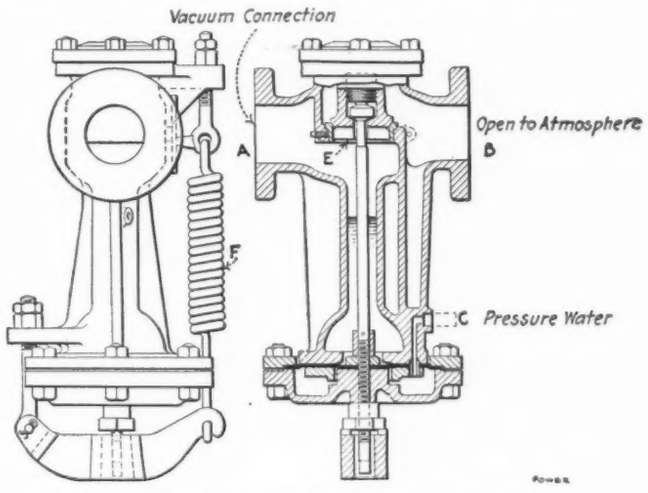
For the fixed charges on the mechanical equipment, the same percentages are allowed. Thus, 12½ per cent. of $46,845 would be $5855, and dividing by the annual output in tons gives $0.535 per ton of ice. The water loss on the cooling tower will be 10 per cent. of 684,000 gal., or 68,400 gal. per day. This, at 10c. per 1000 gal. will cost $6.84 per day, or $0.137 per ton of ice. Summing up, the total cost per ton of ice is as follows:

Water	$0.137
Fixed mechanical charge	0.535
Fuel	0.3096
Operating cost	0.486
Total	$1.4676
Fixed building cost	0.228
Total cost	$1.6956

From the above comparison there is a difference of nearly 4c. per ton in favor of the absorption plant. However, it is only fair to point out that the absorption machine actually uses more heat than the compression machine, but gains in efficiency because of the inefficiency of the steam engine. It is thus apparent that compounding the engine in the compression plant under discussion would better the economy and reduce the cost per ton to approximately that of the absorption plant.

Auld Vacuum Breaker

A new vacuum breaker, known as the Auld, is shown herewith. It is used in connection with jet condensers, especially if the condenser is located on a higher level and a considerable distance removed from the exhaust outlet of the turbine or engine. The vacuum breaker, manufactured by Schutte & Koerting Co., Twelfth and Thompson Sts., Philadelphia, Penn., is connected on or near the exhaust outlet of the machine by its below-seat port *A*, the above-seat port *B* being open to the atmosphere.



Details of Auld Vacuum Breaker

A connection is made at *C* with the circulating-water inlet at the condenser; the operation is as follows:

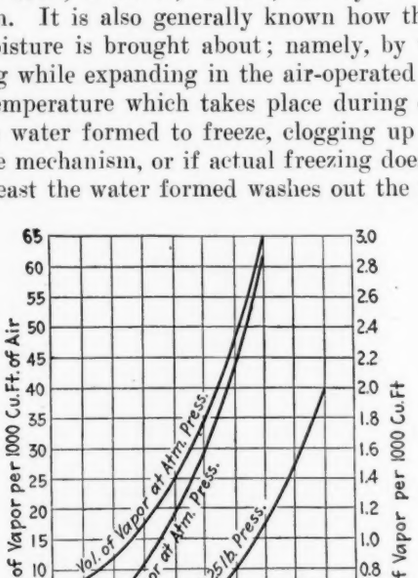
As the pressure of the injection water, to maintain the proper vacuum and insure efficient operation of the condenser, must not go below a fixed minimum (approximately 10 lb. per sq.in.), the vacuum breaker is so adjusted that a reduction of more than 2 lb. in the water pressure will open the valve. This is accomplished by water entering through the inlet *C*, which is fitted to the under side of the diaphragm, thus holding the valve shut against the opening action of the adjustable spring *F*. Therefore, if the pressure of the water is reduced the spring will overbalance the pressure exerted below the diaphragm, thus opening the valve and destroying the vacuum.

Moisture in Compressed Air

By A. Hofmann

SYNOPSIS—Some specific examples showing the amount of moisture present at different temperatures and pressures, the importance of cooling the air, the loss of work caused by moisture and the proportion of cooling water it requires.

Every user of compressed air knows that moisture in air is to be avoided, if possible, when the air is used in pneumatic tools, drills or, in fact, in any reciprocating mechanism. It is also generally known how the trouble due to moisture is brought about; namely, by the vapor condensing while expanding in the air-operated tool. The drop in temperature which takes place during expansion causes the water formed to freeze, clogging up the passages of the mechanism, or if actual freezing does not take place, at least the water formed washes out the lubricant,


Cu. Ft. of Vapor per 1000 Cu.Ft. of Air
Lb. of Vapor per 1000 Cu.Ft
Vol. of Vapor at Atm. Press.
Wt. of Vapor at Atm. Press.
Wt. of Vapor at 25 lb. Press.
Wt. of Vapor at 100 lb. Press.
Temperature, Deg. Fahr.
POWER

Vapor in Air at Different Pressures and Temperatures

causing the tool to wear out rapidly. The usual method of eliminating moisture from compressed air is by cooling the air after it leaves the compressor, thus depositing a portion of the moisture before it enters the pipe lines, and one of the objects of this article is to show just how effective an aftercooler is in removing this moisture at different degrees of cooling.

Moisture Removed by Aftercooler

Atmospheric air always contains a certain amount of moisture and this exists as a vapor or steam gas, and when present in the air it behaves in every way like steam; it exerts a pressure, the amount of which depends upon the temperature and the quantity of moisture present. At a given temperature air can hold a fixed maximum weight of moisture and when it contains this maximum amount the air is said to be saturated. Air containing moisture behaves in accordance with Dalton's law on mixture of gases, that is, each gas or vapor in a mixture exerts its own pressure and the total pressure of the mixture is the sum of the pressures exerted by the gases independently. Thus, take atmospheric air at 80 deg. F., if the air is saturated with moisture the pressure exerted by the vapor will be the pressure of saturated steam at a temperature of 80 deg. F., which is according to any steam table 0.505 lb. per sq.in. abs. Therefore, if the total pressure of the atmosphere is 14.7, the pressure exerted by the air must be 14.7 less 0.505, which equals 14.195 lb. per sq.in. If, now, instead of being saturated the air contains only 70 per cent. of the maximum amount it can hold, that is to say, the relative humidity of the air is 70 per cent.; then the pressure exerted by the vapor would have been only

$$0.7 \times 0.505 = 0.3535$$

and the pressure exerted by the air would be

$$14.7 - 0.3535 = 14.3465$$

When the mixture of air and vapor enters a compressor the vapor again behaves like any gas, that is, it is compressed adiabatically with the air. The vapor then becomes in every sense a steam gas and at the end of compression it is highly superheated, and only by reducing its temperature can the vapor be deposited as water. *Vice versa*, when compressed air containing this vapor is used in any mechanism where it is expanded, the vapor or steam gas is expanded adiabatically, and as every engineer knows when steam is expanded in this way a temperature reduction takes place and a portion of the steam is condensed, and it is this condensation which causes so much annoyance in the air-operated tool.

The volume of vapor which any volume of air contains can be calculated from the following simple formula:

$$V_s = \frac{V p_s}{p}$$

where

V_s = Volume of vapor in cubic feet at pressure p,

V = Volume of air and vapor in cubic feet at pressure p;

p = Absolute pressure of air and vapor per square inch; thus at sea level and atmospheric conditions $p = 14.7$;

p_s = Absolute pressure per square inch exerted by the vapor.

Thus, if the mixture is saturated p_s equals the pressure of saturated steam obtained from a steam table at the temperature under consideration.

The weight of vapor present will be $V \times D$, where D is the weight of vapor per cubic foot and if the air is saturated D equals the weight per cubic foot of saturated steam taken from a steam table at the temperature in question.

In the accompanying diagram is shown the volume of vapor in cubic feet which 1000 cu.ft. of air contains at atmospheric pressure when saturated and at different temperatures. Thus, if the temperature of air is 80 deg. F., the pressure exerted by the vapor is 0.505. The volume V_s of the vapor is according to the above formula:

$$\frac{1000 \times 0.505}{14.7} = 34.3 \; cu.ft.$$

And this is the quantity also shown by the diagram.

Curves are also given showing the weight of vapor which 1000 cu.ft. of air at atmospheric pressure, 14.7 lb.

per sq.in., can hold at different temperatures as well as the vapor which 1000 cu.ft. of free air can contain at 25 lb. gage pressure and 100 lb. gage pressure per square inch. Thus at 60 deg. F. the weight of vapor is 0.8 lb. at atmospheric pressure and at 80 deg. F. it is nearly 1.6 lb. or twice that at 60 deg. From these diagrams it can be determined exactly how much vapor remains in the air after being compressed and then cooled in an aftercooler. For instance, if air is taken into a compressor at 80 deg. F. and if it is saturated with vapor, 1000 cu.ft. of air will contain 1.6 lb. of vapor, and if the humidity is 70 per cent., the amount of vapor will be

$$0.7 \times 1.6 = 1.12 \ lb.$$

If, now, this air is compressed in a two-stage compressor to 100 lb. discharge pressure, the intercooler pressure will be about 25 lb. If water is supplied the intercooler at 60 deg. F., the air will leave the intercooler at about 80 deg., and the amount of vapor 1000 cu.ft. of air at 25 lb. pressure can contain is found from the diagram to be nearly 0.6 lb. In other words, 1.12 lb. less 0.6 equals 0.5 lb. of vapor has been removed by the intercooler, or 45 per cent. of the moisture originally in the air. If, now, the air leaves the compressor at 100 lb. pressure and is cooled in an aftercooler to 80 deg. F., the diagram shows the moisture still in the air to be 0.2 lb., or 18 per cent. of what air was there originally, that is, the intercooler has removed 45 per cent. and the aftercooler 36 per cent. of the vapor.

If instead of cooling the air in the aftercooler down to 80 deg. it had been cooled down to 60 deg., the vapor remaining would only have been 0.1 lb., or 9 per cent., of what it originally contained, and this shows the great importance of cooling the air down to the lowest possible temperature. In the case just cited, by cooling 20 deg. lower, that is, from 80 to 60 deg., one-half of the moisture the air contained was eliminated. Take another example. If the air enters the compressor at 60 deg. F. and 70 per cent. humidity it will contain about 0.56 lb. of moisture, and if on account of insufficient cooling surface the air can be cooled down to only 110 deg. in the aftercooler, practically all of the moisture originally in the air will remain after it has been compressed to 100 lb.

In order then to get all but a small percentage of the vapor out of the air it must be cooled to a low temperature. It is impossible to remove all of the moisture and if the air is used at the same temperature as it leaves the cooler, it will be saturated with moisture and water will deposit when expansion takes place in the tools. To entirely prevent the deposit of moisture, the air must be cooled to a much lower temperature than that which it enters the tools, or what amounts to the same thing, the air must first be cooled and then reheated before it is used.

Moisture Causes Loss of Work

There is another phase of this question to be considered. That is, how does the moisture in the air affect the work done by the compressor? It has already been stated that the vapor must be compressed the same as the air; therefore, the work required to compress the vapor must be considered as lost work since the vapor is eventually practically all deposited as water in the intercooler, aftercooler or pipe lines, and does no work in the tools. For instance, with 1000 cu.ft. of air entering the compressor at 80 deg. F. and 70 per cent. humidity, as before, the vapor will occupy a volume of 34.3 cu.ft. at saturation or

$$0.7 \times 34.3 = 24 \ cu.ft.$$

at 70 per cent. humidity. This is equal to 2.4 per cent. of the total volume of 1000 cu.ft. In other words, the volumetric loss due to vapor is 2.4 per cent. If compression takes place in one stage to 100 lb., the work required to compress 24 cu.ft. of vapor will also be 2.4 per cent. of the total work, or 2.4 per cent. represents the work lost in compressing the vapor. If compression takes place in two stages, the loss is a little less because the vapor that is removed by the intercooler does not have to be compressed in the high-pressure cylinder. Calculation shows in the above case for two-stage compression that the lost work will be about 2 per cent. If the air enters the compressor at 100 deg. F. and 80 per cent. humidity, the same reasoning shows that with single-stage compression the lost work is about 5 per cent. and with two-stage compression about 4 per cent.

This is a loss in compression of air ordinarily not considered. It is true that most users of air compressors know the importance of taking air into the compressor at as low a temperature as possible, and the reasons, of course, are that by so doing a greater weight of air is compressed with a given amount of power expended. In addition to this the air should be taken into the compressor as cold as possible so as to reduce the lost work due to compressing the vapor. For instance, if air is taken from a warm engine room close to 100 deg. temperature and laden with moisture, the figures show that the loss due to compressing moisture may be as high as 5 or 6 per cent. of the total work done.

Cooling Water Required by Water Vapor

Finally, there is one other question to be considered in connection with moisture in the air; that is, what percentage of the total water supplied to the intercooler or aftercooler is used to cool and extract the vapor in the air. Suppose, as before, the air enters the compressor at 80 deg. F. and 70 per cent. humidity, then if the temperature of the air leaving the low-pressure cylinder in a two-stage compressor is 250 deg., and the air is cooled down to 80 deg. F. in the intercooler, the volume of air and vapor being 1000 cu.ft., the volume of vapor alone is 24 and the volume of air is 976 cu.ft. The weight of this air will be 71.8 lb. and the heat extracted from the air will be the weight multiplied by its specific heat and by the reduction of temperature; that is,

$$71.8 \times 0.237 \ (250 - 80) = 2893 \ B.t.u.$$

The weight of the vapor contained in the air as before is 1.12 lb. This must be cooled from 250 to 80 deg., and as the specific heat of the vapor is about 0.46, the heat extracted will be

$$1.12 \ (250 - 80) \ 0.46 = 87 \ B.t.u.$$

To this must be added the latent heat of the 0.5 lb. of vapor which is actually deposited in the cooler. The latent heat of steam at 0.5 lb. abs. pressure is 1046, which, multiplied by 0.5 gives 523 B.t.u. The total heat, therefore, given up by the vapor will be

$$87 + 523 = 610 \ B.t.u.$$

and the entire heat removed by the water in the cooler contained both in the air and the vapor will be

$$2893 + 610 = 3503 \ B.t.u.$$

In other words, of the total heat extracted 610 B.t.u., or 17.4 per cent., is taken from the vapor in the air; namely, 17.4 per cent. of the water used is required by the vapor—a most surprising result.

Specifications for the Purchase of Oil

By J. J. McIntosh

SYNOPSIS—General specifications for the purchase of fuel oil as used by the government and a typical example of the type of contract used by many plants.

Every plant burning oil should incorporate in the contract the amount of impurities permissible in the oil shipments, and every shipment should be tested to see that the forms of the contract are being lived up to. It is a fact that many plants pay no attention to the quality of the fuel they are receiving, and, as a result of their negligence, many of them experience considerable trouble and poor economy in the burning of oil as fuel. Oil as taken from the wells must be properly treated to eliminate the water, gravel and other débris which it contains. In many cases the oil as received at the plants contains large percentages of impurities. Unless the oil is being supplied under a proper form of contract, and each and every shipment tested, these impurities are not only causing trouble and poor economy, but they are being paid for the same as fuel.

The above was forcibly brought home to the manager of a certain company operating in the West. In this instance the form of contract was similar to the one (hereinafter shown) between the city of Phoenix and the Union Oil Co. of California. Unfortunately, this company did not test its oil shipments for nearly a year after operating under this contract. Their operating engineer finally persuaded them to supply him with a reliable oil thief and apparatus to test for impurities. At the end of the first six months after testing their oil shipments, they received a rebate of $2675.

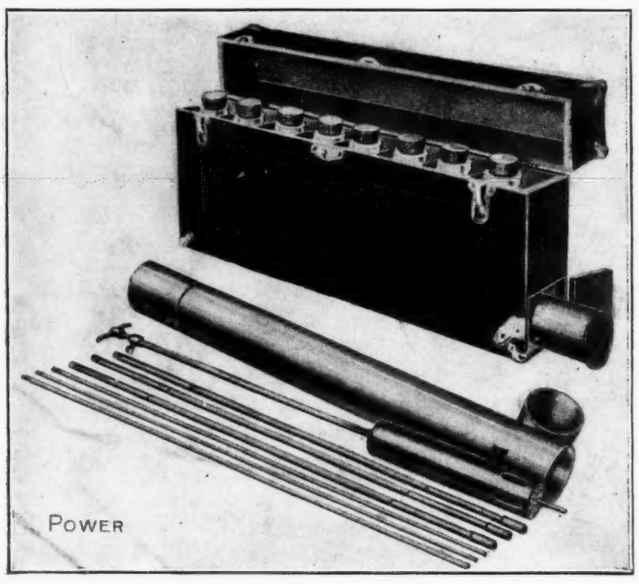


Fig. 1. Type of Oil Thief for Taking Samples

Testing of fuel oil for water and impurities is comparatively a simple matter. The operating engineer can, with a little study, carry on these tests himself and thus make himself more valuable to his company. It must be remembered that fuel oil (unlike coal) shipments received from the same field have, generally speaking, a nearly uniform heat value when freed from water and débris. It is not necessary, therefore, to carry on tests to determine its heat value only insofar as it contains the impurities mentioned above.

In Fig. 1 is shown a type of oil thief for taking samples from oil shipments for testing purposes. In Fig. 2 is shown an apparatus for testing oil for b.s. (brown sediment) and water. Each of the four metal tubes shown in the illustration contains a glass tube graduated in cubic centimeters. The samples to be tested are placed in the tubes and an equal portion of gasoline is added. The rotating element is then revolved at the proper speed for about 15 min. When the tubes are removed from the apparatus there is a well defined line between b.s., water and oil. In this way the percentage of b.s. and water in the oil shipment is easily determined. The above pieces of apparatus are being used to a considerable extent by operating engineers of oil-burning plants.

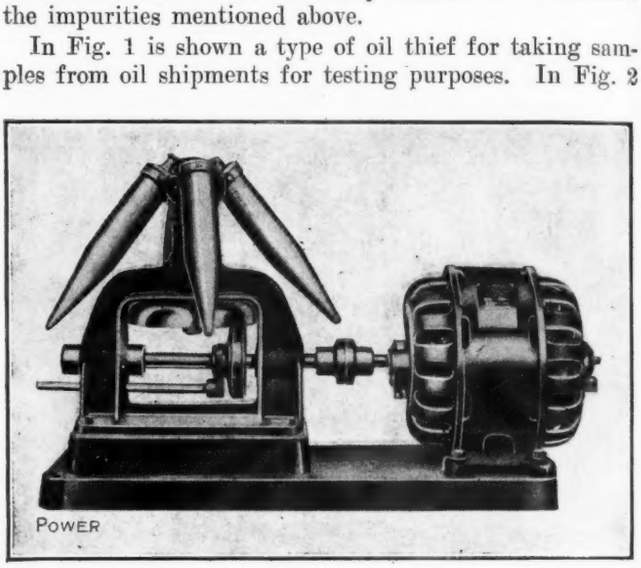


Fig. 2. Apparatus to Test Oil for Brown Sediment and Water

Specifications for the purchase of fuel oil become more rigid each year. Up to within a few years ago oil was bought with little or no regard to the amount of water or other impurities it might contain. A fuel-oil agreement that called for the necessary number of barrels and the price per barrel was considered by the purchaser as being all that could be desired. Below are given the general specifications for the purchase of fuel oil as used by the government:

(1) In determining the award of a contract, consideration will be given to the quality of the fuel offered by the bidders, as well as the price, and should it appear to be to the best interest of the government to award a contract at a higher price than that named in the lowest bid or bids received, the contract will be so awarded.

(2) Fuel oil should be either a natural homogeneous, or a homogeneous residual from a natural oil; if the latter, all constituents having a low flash point should have been removed by distillation; it should not be composed of a light oil and a heavy residue mixed in such proportions as to give the density desired.

(3) It should not have been distilled at a temperature high enough to burn it, nor at a temperature so high that flecks of carbonaceous matter begin to separate.

(4) It should not flash below 60 deg. C. or 140 deg. F., in a closed Abel-Pensky or Pensky-Martins tester.

(5) The specific gravity should range from 0.85 to 0.96 at 15 deg. C. or 59 deg. F.; the oil should be rejected if its specific gravity is above 0.97 at that temperature.

(6) It should be mobile, free from solid or semi-solid bodies, and should flow readily, at ordinary atmospheric temperatures and under a head of 1 ft. of oil, through a 4-in. pipe 10 ft. in length.

(7) It should not congeal nor become too sluggish to flow at 0 deg. C. or 32 deg. F.

(8) It should have a calorific value of not less than 10,000 calories per gram (18,000 B.t.u. per lb.); 10,250 calories to be standard. A bonus is to be paid or a penalty deducted according to the method under section 21, as the fuel oil delivered is above or below the standard.

(9) It should be rejected if it contains more than 2 per cent. water.

(10) It should be rejected if it contains more than 1 per cent. sulphur.

(11) It should not contain more than a trace of sand, clay or dirt.

(12) Each bidder must submit an accurate statement regarding the fuel he proposes to furnish. This statement must show:

(a) The commercial name of the oil.

(b) The name or designation of the field from which the oil is obtained.

(c) Whether the oil is a crude oil, a refinery residue, or a distillate.

(d) The name and location of the refinery, if the oil has been refined at all.

(13) The fuel oil is to be delivered f.o.b. cars or vessels, according to the manner of shipment, at such places, at such time, and in such quantities as may be required, during the fiscal year ending. . . . . .

(14) Should the contractor, for any reason, fail to comply with a written order to make delivery, the Government is to be at liberty to buy oil in the open market and charge against the contractor any excess of price, above the contract price, of the fuel oil so purchased.

(19) A contract entered under the terms of these specifications shall not be binding if, as the result of a practical service test of reasonable duration, the fuel oil fails to give satisfactory results.

(20) It is understood that the fuel oil delivered during the term of the contract shall be of the quality specified. The frequent or continued failure of the contractor to deliver oil of the specified quality will be considered sufficient cause for the cancellation of the contract.

(21) Payment for deliveries will be made on the basis of the price named in the proposal for the fuel oil corrected for variations in heat value, as shown by analysis, above or below the standard fixed by the contractor. This correction is a pro rata one and the price is to be determined by the following formula:

$$\frac{\text{delivered calories per gram (or B.t.u. per lb.)} \times \text{contract price}}{\text{standard calories per gram (or B.t.u. per lb.)}} = \text{price to be paid}$$

Water that accumulates in the receiving tank will be drawn off and measured periodically. Proper reduction will be made by subtracting the weight of the water from the weight of the oil deliveries.

The specifications given above provide for the purchase of fuel oil by weight. As such oil is frequently delivered by volume, it is important to note the temperature of a delivery and to allow for the expansion due to this temperature when computing the weight of the delivery from the volume. From the volume of the oil at the temperature of delivery, the volume at standard temperature, 15 deg. C., or 59 deg. F., should be computed in the manner given below.

The coefficient of expansion of ordinary fuel-oil residue, of asphaltic base, is approximately 0.0006 per 1 deg. C.

Hence, if the temperature of the delivery is above 15 deg. C., then

$$(N \text{ deg. C.} - 15 \text{ deg. C.}) \times 0.0006 = \textit{Correction}$$

This correction is to be added to the specific gravity at the higher temperature to give the standard specific gravity at 15 deg. C.

If the temperature of the oil delivered is below 15 deg. C., the correction is to be subtracted from the specific gravity at 15 deg. C.

Since a gallon of water at a temperature of 15 deg. C. weighs 8.3316 lb., the weight in pounds of a gallon of oil at 15 deg. C. is 8.3316 times the specific gravity of the oil at that temperature.

Similarily, since a cubic foot of water at 15 deg. C. weighs 62.3425 lb., the weight in pounds of a cubic foot of oil at 15 deg. C. is 62.3425 times its specific gravity at that temperature.

Following are given the principal items from a fuel-oil agreement in force between a railroad company and a large oil company, both of whom are operating in the Southwest:

(1) Oil to be crude petroleum in its crude or natural state, or the product of crude petroleum, distilled or reduced.

(2) Gravity of oil to be between 13 and 26 deg. Baumé at 60 deg. F.

(3) Oil must not contain water, sand or other foreign matter in excess of 2 per cent.

(4) Oil to be paid for on the basis of volume at 60 deg. F. after deducting the percentage of water and foreign matter as may be shown by test outlined below.

(5) Samples for testing to be taken by means of a car thief from each car load, or fraction thereof. Oil thus obtained to be tested for brown sediment, sand and water. The percentage of brown sediment, sand and water will be determined by the gasoline test.

(6) Should the oil tested contain in excess of the specified amount of water and other impurities, the shipment represented by it will not be accepted, but will be returned to the owners, who shall be responsible for the freight charges, both ways, on the rejected shipment.

To show the different contracts which are in use for the purchase of fuel oil the agreement now in force between the water-works of the city of Phoenix and one of the largest oil companies operating in California is given in the following. It is a typical example of the type of contract a great many plants are working under:

This agreement, made in duplicate this eleventh (11th) day of January, 1910, between Union Oil Company of California, a corporation, party of the first part, and the Common Council of the City of Phœnix, Arizona, hereinafter called the party of the second part.

Witnessed, that in consideration of the mutual covenants hereinafter contained, the party of the first part agrees to sell and deliver, and the party of the second part agrees to use oil as fuel at Phœnix, Arizona, and to purchase, take and receive from said party of the first part all the fuel oil that said party of the second may require for its own consumption at the place, during the time and at the price or prices hereinafter mentioned, to-wit:

Quantity: The party of the first part shall not be compelled to deliver to the party of the second part, during the term of this agreement, more than one thousand barrels of forty-two (42) gallons each in any one calendar month, but the party of the second part agrees not to buy any fuel oil from any other person, firm or corporation other than the party of the first part, so long as the party of the first part is ready, willing and able to supply all the fuel oil requirements of said party of the second part, on the terms herein specified.

Quality: Said fuel oil shall not be heavier than fourteen degrees Baumé at 60 deg. F. If said oil shall contain water and sand in excess of one per cent., as shown by the gasoline test, then said party of the second part shall be entitled to such excess at the price herein mentioned, provided claim therefor is made within ten days after delivery thereof.

Point of delivery: Said fuel oil shall be delivered at Phœnix, Arizona.

Said party of the second part agrees to give ten days' notice of its said requirements, and to provide suitable and sufficient tanks to promptly receive and store said oil.

Price: The price or prices to be paid for said fuel oil per barrel of forty-two (42) gallons each, shall be as follows: One dollar and sixty-five cents per barrel ($1.65); based on present freight rates of $6 per ton. If freight rates should increase or decrease during the life of this contract, then

such increase or decrease of freight shall be added to or deducted from the above price of oil per barrel delivered as hereinbefore specified.

Times of Payment: All deliveries hereunder in any calendar month shall be settled and paid for in United States gold coin, at the price or prices aforesaid, not later than the tenth day of the next succeeding month. Default in any payment for a period of ten days shall justify the cancellation of this contract, but if said contract is not canceled therefor, and while such delinquency continues, the party of the first part may require payment on all deliveries made.

Liability: It is further mutually agreed that neither of the parties hereto shall be liable for any delays or damages occasioned by, or arising from strikes, or other labor disturbances, earthquakes, fire, action of the elements, war, insurrection, riot or rebellion, or interference by civil authorities; and the party of the first is not to be liable in the event of failure of railroad, or other transportation, not controlled by it, nor in the event of its inability to procure tank cars from the railroad company for making deliveries hereunder.

Should party of the second part sell or dispose of its plant or should the same cease to be operated for a period of ninety days continuously, the party of the first part shall have the right to cancel this contract, by giving said second party written notice of its election so to do.

Progress on Panama-Pacific Exposition

Construction work on the numerous large wooden buildings for the Panama-Pacific International Exposition at San Francisco is making rapid progress. Among the principal buildings already erected are the Palace of Machinery, covering three acres, the Fire House, the Education Building, the passenger station, and warehouses, ferry slips and oil houses. The framework has been started for the buildings for Mines and Metallurgy, Various Industries, Manufactures, Liberal Arts, Food Products and Agriculture.

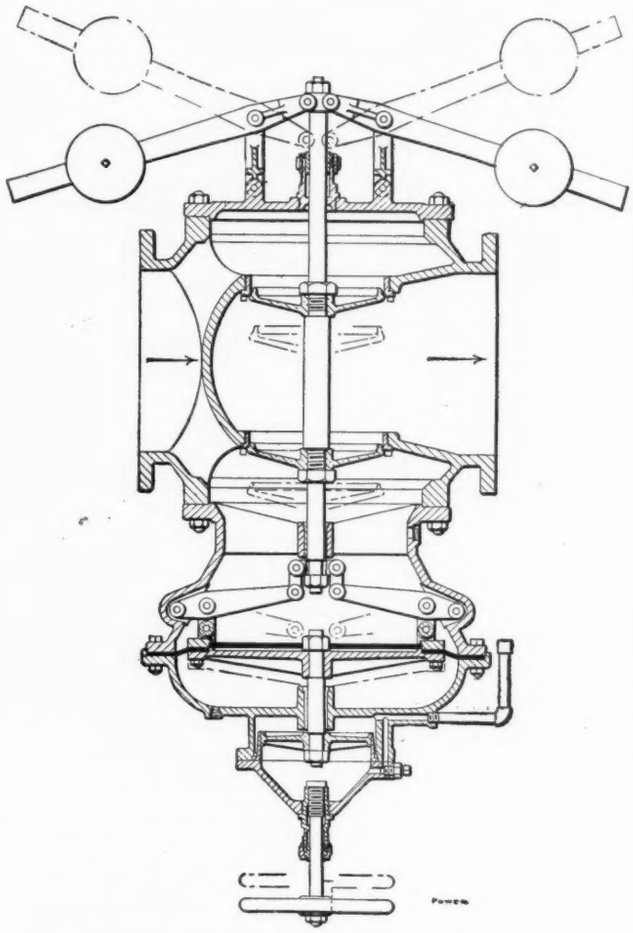
Thirty-five states and territories and certain of the foreign governments have announced their intentions to secure space; 177 congresses and conventions have been arranged to meet in San Francisco during the fair, and an attendance of from 10,000,000 to 18,000,000 is anticipated for the exposition. It is said that neither Germany nor England will take exhibit space.

The cost of the exposition is estimated at $80,000,000, of which the state of California and the city of San Francisco will each pay $5,000,000, San Francisco subscribes $7,500,000 and individual exhibitors $25,000,000.

Schutte Automatic Flow-Regulating Valve

This flow-regulating valve, illustrated herewith, is used on the low-pressure steam supply to mixed-pressure turbines which operate on an intermittent supply of exhaust steam from engines, pumps, etc., and partially on high-pressure steam through a separate valve when the low-pressure supply is cut off. The function of this valve is to maintain a steady pressure to the turbine of 16 to 17 lb. absolute and to close when the pressure drops below 15 lb. absolute, due to the insufficient supply of exhaust steam from the engines. The object is to prevent the loss of vacuum and consequently reduced efficiency of the turbine.

This valve, which is manufactured by the Schutte & Koerting Co., Twelfth and Thompson Sts., Philadelphia, Penn., as the section shows, is of the double-seat type, globe body, with a water chamber below which is in communication with the inlet side of the body, at the bottom of which is a large rubber diaphragm, constantly submerged and thus protected against the heat of the steam. This diaphragm is connected to the valve spindle by means of multiplying levers so that the motion of the diaphragm is relatively small as compared with the movement of the valve disks. A dashpot is fitted below the diaphragm to absorb sudden shocks and in the bottom cover of this a screw spindle with handwheel is placed, by means of which the disks can be forced to the seat for closing the valve. The spindle passes through a stuffing-box in the upper cover above which it attaches to the balancing levers fitted with adjustable weights.



SECTION THROUGH FLOW-REGULATING VALVE

The adjustment is such that any increase in pressure over zero gage will open the valve and any reduction in pressure below zero will close it. A hydraulic closing cylinder can be fitted at the top when so desired for closing the valve from a distance by means of pressure water, controlled by a three-way cock. This valve can only be built in the globe form, as shown.

To Develop 52,000 Hp.—Another hydro-electric development planned on Cheat River in West Virginia is that of the West Virginia Development Co., This water power is at the Beaver Hole site, about 13 miles from the Pennsylvania state line, and is estimated to develop about 52,000 hp. for use in West Virginia and southwestern Pennsylvania markets. The company has not determined upon definite plans for beginning construction, but proposes a dam about 1200 ft. long and 100 ft. high, with the power house built in and forming part of the dam.

Editorials

Control of the Water Powers

If the price of eggs goes up to eight or ten cents a piece, one may go without them; if your shoeblack or barber doubles the price of his services, you may shine and shave yourself. If the electric company persists in charging much more for its service than it is considered worth, a community may, in its corporate capacity, make the current itself and distribute it at cost. The best defense on the part of an individual or of the people collectively against extortion is the ability to do for themselves.

But there are some things which the people think they cannot do, and under present circumstances probably cannot do, short of an industrial revolution, for themselves. They will continue to pay eight or ten dollars a ton for coal that they used to buy for three, because a band of industrial barons had the "thrift, enterprise and industry" to acquire most of the workable deposits and dominate the transportation and distribution of mined coal.

Is this process to be repeated with the water powers?

With the increasing price of fuel, both coal and oil, and especially with the electrical development which makes their power transmissible to points of application remote from its source, the value of water powers is increasing by leaps and bounds. The increasing uses of power for interurban trolleys, farm work, outlying industries, to say nothing of the growing urban demands, promise a rapid absorption of the power available in our rivers as it is developed.

What are to be the conditions of this development?

It is desirable that they should be developed, if the people are thereby put in the enjoyment of light and power at a decreased cost. Otherwise they had better remain in the people's hands as an available asset against the time when they are driven to the conviction that they can do things on their own account.

The water rights of most of these rivers remaining still in the hands of the people and cheap power being one of the most desirable and beneficient things that a people can have, it would seem to be the natural and logical thing for the people in their collective capacity, that is the government, which is of the people, to develop them and to furnish electrical energy to themselves at cost, and if this is to be done at cost it would not seem to matter materially whether the development and administration were undertaken by the federal government or that of the state in which the site were located, except as to the control of patronage.

If the power privileges are to be conceded to private interests which will undertake their development only with the obvious purpose of making money out of it the fact should not be lost sight of that the money made will come out of the users of the power; and provision should be made that this should not exceed a generous but fair profit upon the capital actually invested. There is no sense in giving a water power away that it may be developed and power from it sold at a price which will just compete with steam-made power when by the exercise of a legitimate governmental activity the people can be put into perpetual enjoyment of that power at the cost of its production.

At the recent conservation congress, Gifford Pinchot declared that concentration in water-power control had doubled in ten years and that the ten leading interests today control sixty-five per cent. of all the developed water powers of the United States. A warp is put into the argument that these concessions must be given to big industrial interests in order that they may be developed at all by the fact that during the past five years the concentration of control by ten large financial groups has increased seven times as fast as the power developed by these groups, and for the past two years their control of undeveloped power has increased twice as fast as that developed by them.

The states' rights issue over which the congress split seems to have been injected by the interests which prefer the diversified and more manageable control of the state legislatures to the potent and unified policy and control of the central government.

Air of Infiltration

Much progress has been made in steam-boiler economy, especially along the lines of automatic stokers, improved grates, baffling, distribution of boiler tubes, circulation within the boiler, flexible draft control, automatic recording apparatus, water treatment, etc. All these have tended to raise the degree of efficiency attained in the generation of steam.

In fact, practically every factor affecting the proper handling of fuel has, with one exception, received careful and more or less constant attention. This is the housing of the apparatus which is expected to absorb the heat energy produced in the furnaces. The boiler setting, the mass of brickwork which surrounds every vital part of the steam-producing apparatus, is built with the primary object of confining the heat and forcing it to come into intimate contact with every part of the apparatus capable of heat absorption. For this reason, the settings are lined with firebrick so that great torrents of heat-laden gases may be forced through the mass of tubes. The heat insulation of the firebrick is increased by an outer coat of heat-resisting common or red brick. In short, considerable effort is made to keep all of the heat energy within the setting.

In most cases, however, but little effort is made to keep the cool air of the boiler room from entering the heat-absorbing apparatus and so reducing the amount of heat available.

How many engineers know whether their plant is running at a lowered efficiency, because of the cooling effect which air (drawn in through the brickwork and around poorly hung clean-out doors, etc.) exerts upon the heat-absorbing surfaces?

How many are supplied with the proper apparatus for

carrying out simple tests, to satisfy themselves that this loss is minimum or of such magnitude that they cannot well afford to let it continue?

The air from the boiler room, drawn into the setting of boilers by the action of the reduced pressure within those settings, is commonly called air of infiltration, because, as the name implies, it is generally admitted through the settings by a process of seepage following the fine interstices in the bricks, the joints between the bricks, and around clean-out doors, and door frames, etc. It is only through tests that its presence can be known, and it is well worth while to make these tests from time to time as the article on page 72 very forcibly shows.

The settings must be watched, frequent tests made and, when required, a general overhauling given because in most plants the settings are subjected to temperature changes which are likely to work havoc with the best of brickwork. This is especially true in plants where much banking of fires is essential, such as railway power stations, during periods of base load. At the expiration of the base-load periods and just preceding peak loads, the banked fires are raised to full intensity in the shortest possible time. Such severe changes cause more damage to the average setting than long periods of running at full intensity of fires but at an almost uniform temperature.

Boiler Settings and Efficiency

To obtain good operating results the equipment must be of proper design to start with. It would not be reasonable to expect 80 per cent. efficiency from a boiler unit designed for anthracite coal but fired with bituminous. Yet, it would be astonishing to learn the number of misfit boiler installations that have been and are being placed in service, often on the recommendation of men who should know better.

Much of this misapplication is caused by human conservatism, unwillingness to change the ancient order of things. A company puts in a boiler of a certain type and set in a certain manner, usually conventional, in its Eastern plant and secures satisfactory results. Later, it establishes a Western branch and, often, the man with the buying authority, be he the owner, the consulting engineer or the operating engineer, confidently orders equipment as nearly as possible in duplication of that in the Eastern establishment. The plant in the East made good, he sees no reason why the new one in the West will not.

The makers of apparatus themselves sometimes make the similar mistake of recommending something that has proved successful elsewhere for conditions that are not at all the same.

The designing of a boiler plant involves many factors. Laying out the setting alone is a study and almost each installation is a law unto itself. The character of the duty required and the fuel to be employed are the fundamental considerations. These should determine the kind of grate and firing to employ—manual or mechanical stoking. High rates of driving and high load factors usually indicate automatic stokers. In some localities, labor conditions and cost may be important. The cost of coal has its influence. When the method of stoking has been selected the type of grate or stoker must be determined upon. Here, further study is required, because numerous distinct types all possess some merit, yet probably only one is best suited to the case in hand.

Next, will come the type and dimensions of the furnace, then, the gas passages and the baffling. On these points, authorities that otherwise agree, are prone to differ greatly. In many quarters the combustion arch—so long regarded as the panacea for the smoke evil in the burning of soft coal—is now falling into discredit. Latest thought indicates that the draft is the thing. And so, they are reducing the size of the furnace arch, and moving it toward the rear of the setting, simplifying and amplifying parts of the gas channel and insisting that adequate draft capacity be provided. Results have indicated that such changes are often beneficial. The change in arch location and design, at least, results in reduced cost of maintenance.

The practice with regard to baffles has also seen change recently. It is still more or less orthodox to compel the gases to follow a tortuous path by baffling, on the theory that only great linear contact at high speed is conducive to economy. The exact effect of velocity on the efficiency and rate of heat transmission is still far from being well understood. It would seem that there is one velocity, probably neither the highest nor lowest practicable, at which efficiency is maximum for any given installation. This velocity is probably different for different operating conditions and must be obtained by proper baffling and draft adjustment for the individual case. A standard baffle practice for all applications is not conducive to highest efficiency from the standpoint of design.

Beginning with this issue of POWER, there will be a change in the makeup of the paper, in that subjects along special lines will not be grouped under departmental headings as heretofore. This does not mean that we intend to slight the electrical, gas-power, refrigerating and heating and ventilating fields, as those of our readers who are specially interested in these branches will find just as much of this material as formerly. A short time ago the two letter departments were consolidated. Now with them will be put all letters formerly found in the various departments, and the heading will be "Correspondence."

Both of these changes affect the arrangement only.

The result of the inspection of locomotives by the Interstate Commerce Commission, instituted by the Act of Congress of Feb. 17, 1911, is already apparent in reduced injuries and loss of life. In 1912 there were 856 accidents resulting from failures of locomotive boilers and their appurtenances, causing 91 deaths and injuring 1005 persons.

In 1913 these items were reduced to 820 accidents, 36 deaths and 911 injuries. Of 90,346 locomotives inspected, over half (54,522) were found to be defective, and 6690 were either held out of service or ordered changed and strengthened to conform to the requirements of the law. It is difficult to understand the attitude of certain railroad officials who object to making some of the repairs and revisions called for, although the defects are such as might cause serious accident, on the ground that the law does not cover such defects and that the commission was exceeding its authority in pointing them out and calling for their correction.

Correspondence

Convex Boiler Heads

A recent disastrous boiler explosion, and several similar accidents, where the cause of the trouble was ascribed to so called breathing action in the turn of the flange of convex drum heads, point to the necessity of greater care in designing these heads. The usual way of determining the strength of these heads is to consider them part of a sphere, according to the formula:

$$P = \frac{t \times T.S.}{\frac{1}{2} R \times F}$$

in which

P = Allowable pressure;
t = Thickness of plate in inches;
$T.S.$ = Tensile strength of material in pounds per square inch;
R = Radius of curvature;
F = Factor of safety.

The different state boiler laws that have made their appearance of late years stipulate a minimum factor of safety of five to be used in connection with the foregoing formula for convex heads without manholes, which would seem ample, although it is quite common to find the factor of safety greater than five on the heads at the allowable pressure as determined from the strength of other parts of the same boiler.

It would appear from past behavior of these heads that some important factor in their design has in cases received rather scant attention, and this in the writer's opinion is the radius at which the flanges are turned over.

With some exceptions, it is practically an established practice to bump the heads to a radius equal to the diameter of the drum on which the heads are to serve. No such uniformity of practice appears to exist in regard to the radius of the turn of the flange. This radius one may find to range anywhere between the equivalent of five or more times the thickness of the plate to almost nothing, or what practically amounts to a sharp corner.

It is obvious that there ought to be a reasonably good ratio for this radius in relation to the thickness of plate and the radius of curvature of the heads. In analyzing the character of the strains that are to be anticipated, one may expect, as is seen from Fig. 1, that the drum head when subjected to internal pressure tends to assume the shape of a true half sphere as in the dotted lines, which would be the ideal form of convex head and in which the strains due to internal pressure would be uniformly distributed.

In reality the convex head actually does flex outward, Fig. 2, though to a very small extent. The part that has to bear the brunt of this flexing is the heel of the flange. It follows that the smaller the radius at which the heel of the flange is turned over, the more localized will be the strains at one point (Fig. 2), and the construction becomes inferior to that of a well braced flat head. The result is, that after a long period of repeated straining and subsequent relaxing, the metal begins to crack.

Corrosion will naturally be most active on the clean metal thus exposed in the cracks and aids materially in weakening the sheet, which eventually may rupture.

Thus it will be seen, that the true factor of safety on this form of head is likely to be a very uncertain value, if the turn of the flange is not a sufficiently gradual one, and a factor of safety of five in connection with the thickness of the head may not be sufficient to take care of

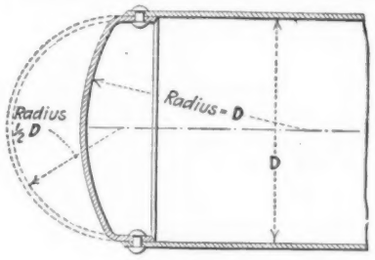


Fig. 1

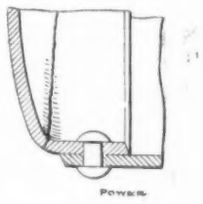


Fig. 2

Showing Tendency of Head to Assume Spherical Shape and Section Where Cracks Are Most Likely to Occur

structural weakness due to too sharp a turn of the flange.

The establishing of a minimum radius, say, of not less than five times the thickness of plate, undoubtedly would prove of benefit in the prevention of explosions.

Hartford, Conn. H. J. Vander Eb.

[The contention that the weak point is the turn of the flange, is borne out by the diagram of the stresses in a boiler drum, on page 50.—Editor.]

Spheres and Convex Heads

In considering the strength of a thin hollow sphere, to resist internal pressure, the simplest method is to divide the vessel into two equal parts, or hemispheres. Bearing in mind the laws relating to thin hollow cylinders, consider the ends to be truly spherical. By such procedure we find the unit stress on such ends, heads or hemispheres, to be one-half that on a longitudinal seam of the cylinder. Hence, in calculating the allowable pressure, R the radius is taken at one-half that of the cylinder. On this basis there are two formulas; one to determine the allowable pressure, and the other to find the required thickness for a given pressure; these are as follows:

$$t = \frac{P \times R \times F}{T.S. \times E}$$

$$P = \frac{T.S. \times t \times E}{R \times F}$$

where R equals one-half the radius of the sphere; t, the thickness of the plate; $T.S.$, the tensile strength in pounds per square inch; F, the factor of safety, usually 5; and E, the efficiency of the riveted seam.

Such rules are used in practice for spherical vessels, notably in straw-board mills, which are often built up to 14 ft. in diameter.

Applying the foregoing to convex heads, for steam boilers, the rules declare the head should be bumped or convexed to a radius equal to the diameter of the drum or cylinder of which it forms an end. At present the rules pertaining to such heads are those of the U. S. Government, and the states of Massachusetts and Ohio. By the government rules where P is the pressure allowed; R, one-half the radius to which the head is convexed; t, the thickness of plate and S, one-fifth the tensile strength.

$$P = \frac{t \times S}{R}$$

The Massachusetts rule using the same constants, except taking $T.S.$ at full value and using an apparent factor of safety of 5, is as follows:

$$t = \frac{R \times 5 \times P}{T.S.}$$

By the former the inspector finds the allowable pressure, and by the latter the designer is presumed to find the thickness required. In each the factor of safety is presumed to be five. Comparing these with the rules pertaining to a sphere, it is noted that no value, other than contained in the factor of safety, is taken as respects the efficiency of the seam for the head. Indeed, were the seam double-riveted no more pressure would be allowed, although the strength of the seam would be greatly increased. Inasmuch as these rules are based on the laws of spheres, there exists no good reason why the efficiency of the seam should not have been considered in formulating ways to determine P and t just as in spherical rotary vessels.

But, as a matter of fact, convex heads are spherical only at certain points, and therefore such rules are far from correct. In practice the head must diverge from spherical form where it bends to enter the cylinder, at the point, termed in practice, the heel of the flange, and this point is from 3 to 6 in. from the center of the rivets. In the sphere the laws depend upon the vessel being truly spherical, and in the cylinder as being truly cylindrical, so also as regards convex heads, if we are seeking exactness. It is contended by some that flexure or bending action occurs with variations of pressure in the circular zone at or near the heel of the flange where the spherical lines terminate. Certainly, it is an error to assume that this section is spherical and governed by the laws of a sphere as respects internal pressure.

It is more than a mere coincidence that two recent explosions are on record wherein the initial ruptures occurred by failures of convex heads in the zone referred to. In the first case a Climax boiler exploded in Brooklyn a few years ago. The dimensions as respects the shell, head and pressure were as follows: Diameter of shell, 48 in.; steam pressure, 150 lb. (adding 10 lb. for water head = 160 lb.); original thickness of bottom head, ¾ in. The external side of the head was badly wasted away and the thickness at the point of fracture was about 0.33 in. The diameter of opening was about 41 in. In the second instance a Wickes vertical boiler on Staten Island exploded by reason of a circular section of the bottom convex head failing, tearing out a section of about 80 in. diameter; see POWER, Nov. 4, for details. The data of the latter are as follows: Diameter lower drum, 90 in.; pressure carried, 150 lb. (adding 10 lb. for water head = 160 lb. total). In this case internal corrosion had wasted away the plate to some extent. In the main, the thickness was reduced to $\frac{9}{16}$ in., and, at one section of fracture, to about ½ in. This was measured only at the ring remaining of the head that failed, the other section not being available. Several experts, however, coincide in the measurements being approximately correct in each case.

It is worth while to compare the data submitted with the existing rules, or, rather, laws, for convex heads. Taking the Climax boiler with $t = 0.33$ in., we have, using 60,000-lb. T.S.

$$\frac{0.333 \times 12{,}000}{24} = 166 \textit{ lb. allowable pressure}$$

and with the Wickes head, and same tensile strength

$$\frac{0.5 \times 12{,}000}{45} = 133 \textit{ lb.}$$

The apparent factor of safety in each is 5; hence, the calculated bursting pressure $166 \times 5 = 830$ lb. with the Climax, and with the Wickes, $133 \times 5 = 665$ lb., whereas the heads failed at, or very nearly, the pressures stated above. Evidently the metal, even at the reduced thickness was in a weakened condition or small annular cracks were present as a result of the continued flexure. It seems therefore these two explosions graphically show the true bursting pressure of such heads, and that the factor of safety as given in the rules is a dangerous fallacy.

In the case of the Wickes boiler, it is evident the designer was strictly within the U. S. Government rules, for, with ¾-in. head, 60,000-lb. tensile strength and a diameter of 90 in., we have

$$\frac{0.75 \times 12{,}000}{45} = 200 \textit{ lb. allowable pressure}$$

One would inquire, however, if ¾ in. is ample for a 90-in. head for 200 lb.; also, why the builders of the Climax boiler used the same thickness for a 48-in. head without a manhead. According to the rule, it would have been safe with an apparent factor of safety of 5, using 60,000-lb. tensile strength and 200 lb. pressure, as follows:

$$\frac{24 \times 5 \times 200}{60{,}000} = 0.4 \textit{ inch}$$

or but little more than the thickness at the point of rupture.

Comparing the pressure allowable by the rules with the actual bursting pressures, it appears the real factor of safety is far below what it should be.

Digressing to inquire further concerning these explosions it has been said that water-hammer may have been the cause; that, owing to the vertical design, were the water lifted, and then, in return, projected violently on the bottom head this would have caused or accounted for the explosions. With regard to this feature, so far as could be learned from all available sources, the boilers were in actual service for some time prior to the explosions and hence the conjecture was not borne out nor proved. That convex heads in vertical boilers would be peculiarly weak in this respect is true, and it follows that this feature should be considered in designing such boilers. The 48-in. shell, with ¾-in. head, is conceded to have met such requirements, but, unfortunately, neglected to provide facilities for examination of the head on the outside, and this inaccessible section wasted and rotted away. In regard to the Wickes head, it has been said the initial rupture took place at the point of fracture where corrosion was greatest and then tore completely around the

circle, charging the failure to the thin section alone. In the writer's opinion this would not have been the case for, with merely local weakness at one point, and opening by reason of sheer weakness, would have been similar exactly to the rupture of a bag on the fire-sheet of a horizontal tubular boiler, and, hence, would not have extended in a circular manner for a distance of about 250 in. unless the metal was already greatly weakened by flexure. Of course, in this head, braces were attached for the purpose of assisting the tubes in supporting the flat tube sheet, as the central tubes in this type of boiler are in compression, and the ones nearest the flange in tension. These braces had given trouble; one was broken at the heel. The writer did not, however, consider the braces as a direct cause for, evidently, such was not the case. That corrosion was a contributory cause is, of course, true in each instance.

Returning to the question of the laws pertaining to convex heads it is well known that most builders using convex heads in drums provide much heavier ones than required by the laws, as, indeed, was the case in the Climax boiler. The rule used by such builders may be based on experiments made on such heads, or may be based on good judgment gained in years of actual practice. It is to their credit that they have provided heads far heavier than required by the law; that they have acted on convictions to protect the buyer.

To deduce a better rule than the present laws, it has been necessary merely to measure what is the best practice of today, and this is as follows:

Let

P = Steam pressure allowable;
t = Thickness of plate in inches;
$T.S.$ = Tensile strength (not more than 56,000 lb.);
F = Factor of safety ($8\frac{1}{3}$);
R = One-half the radius to which the head is convexed, the radius to be not less than the diameter of the drum.

With this data, and for heads without manholes, we have

$$P = \frac{t \times T.S.}{R \times 8.33}$$

$$t = \frac{R \times 8.33 \times P}{T.S.}$$

The results obtained by the proposed rule are:

42 in. diameter and 200 lb. pressure, t = 0.625 in.
48 in. diameter and 200 lb. pressure, t = 0.714 in.
90 in. diameter and 200 lb. pressure, t = 1.338 in.

By the U. S. Government, Massachusetts and Ohio rules we have:

42 in. diameter and 200 lb. pressure, t = 0.375 in.
48 in. diameter and 200 lb. pressure, t = 0.4286 in.
90 in. diameter and 200 lb. pressure, t = 0.803 in.

In the foregoing the tensile strength is taken at 56,000 lb. for the reason that mild steel with low carbon content is better material for this purpose than higher carbon steel. Ductility is essential for long life and is perhaps less susceptible to corrosion.

The weakness of the present laws as proved by the heavier heads used, together with evidences obtained from the two explosions, should result in revising the laws about on the lines given above. It is to be noted that in each of these explosions seven lives were lost, and, in addition, several persons were injured, together with heavy property damages. It does not appear proper that this be deferred pending further evidence in the way of disastrous explosions, that is, if we are really desirous of the safety for which ostensibly steam-boiler laws were created. Pending some action to this end, may I suggest the above be carefully considered by the operating engineer and managers of plants, for, possibly, under their care may be badly corroded heads similar to the ones that failed in the cases referred to.

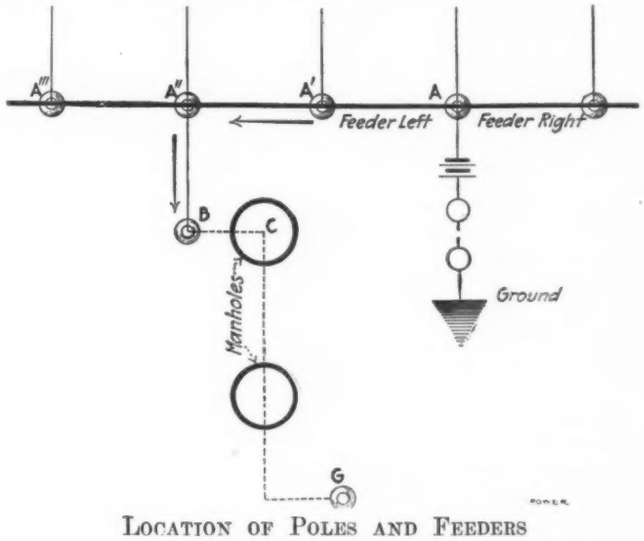
T. T. PARKER.

New York City.

Pocket Compass for Locating Ground

I was once assigned the duty of locating a dead ground upon a 220-volt lighting system, and while the ammeter method as outlined by "Foster's Electrical Engineers' Handbook" could have been employed, a more novel method was resorted to. In the diagram one side of the circuit only is shown, A, A', etc., representing tops of poles from which service wires are tapped from feeders; B is the top of a pole from which the feeders change from aërial to underground lead-covered cable by way of manhole C.

The main generator switch being opened, the positive side of a storage battery, the leads of which were avail-



LOCATION OF POLES AND FEEDERS

able at various points of the system, was connected at A'; the negative side being grounded through a bank of lamps.

As the poles were stepped it was an easy matter to climb them and with a compass determine which path the current was taking to the ground, thence back to the battery. Ascending the pole A' and placing the compass over the feeder to the right, it was not affected, but when placed over the feeder to the left it was affected in the usual way, thus indicating a flow of current. Ascending the poles in order it was observed that the current was confining itself to the feeder as indicated by the arrow. As manhole C was entered, the compass was not affected so strongly by any one conductor, but feebly by several or by their sheaths. This action led to the belief that the bad conductor to which the positive side of the battery was connected was grounded between the top of the pole and the manhole.

To verify this conclusion, the connection at A' was changed to G and pole B was climbed again, from which it was observed that no appreciable current flowed into

any of the aërial leads. Manhole *C* was again entered and here is where I was nonplussed for a few minutes, because the compass pointed in the opposite direction to that which might have been expected. It suddenly dawned upon me that the lead might be grounded to its own sheath, and in such a case, it having been decided that the lead was grounded between the manhole and the top of the pole, nearly all of the current would return through the sheath. Since the conductor was concentric, we should expect the two fields to neutralize each other beyond the detection of a compass of the pocket variety. But in this case the field set up by the current returning through the lead sheath influenced the compass, but, of course, quite feebly as compared with its unneutralized effect.

When starting out to locate the ground I had expected that the dead ground was the outgrowth of some incipient fault in the old work, some new extensions to the system nearing completion not being in the least suspected. But it proved, however, that the fault was in the new cable where it had been spliced onto the aërial line. The lead sheath had not been stripped back far enough and where the underground cable was soldered to the aërial cable the solder had run and connected solidly from copper to sheath.

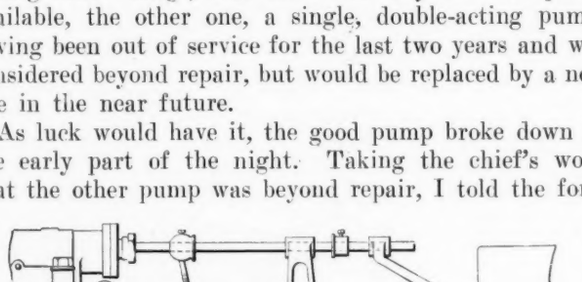
AMOS J. CARR.

Cape Cottage, Maine.

Emergency Repair to Feed Pump

Sometime ago I went to work in a factory as night engineer. As the chief showed me around he mentioned, among other things, that there was only one feed pump available, the other one, a single, double-acting pump, having been out of service for the last two years and was considered beyond repair, but would be replaced by a new one in the near future.

As luck would have it, the good pump broke down in the early part of the night. Taking the chief's word that the other pump was beyond repair, I told the fore-



HOW THE WEDGE WAS APPLIED

man that we would have to shut down. He said that if we did, it would mean a big loss to the company.

I then decided to try the old pump. It refused to go more than one stroke. There was the possibility, however, of keeping it going if somebody stood by and worked the rocking lever.

I made several adjustments to the rocker connection and found that the only trouble was that during years of operation the rocker roller wore down in the diameter to such an extent that it could not lift the rocker at both ends.

I made a quick, emergency repair in a couple of minutes by whittling a piece of wood wedge shape, the thickest end being about ⅛ in., and tied it with two piece of wire to the working face of the rocker, and the pump worked satisfactorily while the other pump was being repaired.

The next day I turned down, in the lathe, another roller of ⅛ in. larger diameter and the pump was as good as new.

VICTOR BONN.

Washington, D. C.

Fitting and Failure of Pistons*

I have read with considerable interest the article in the Nov. 18 issue of POWER, describing a piston failure in one of the mills at Paterson, N. J.

Why was the piston made a taper fit? This alone invites disaster.

Some years ago, when I followed locomotive work, it was the usual practice to make taper fits on both the crosshead and piston. The piston rod was ground into a perfect fit, then forced to place and keyed. Just why a part like a piston, that will in all probability never be taken off until worn out, should be so fitted is beyond me.

When it came within my province to direct such matters the pistons were fitted straight; in fact, the rod was made a few thousandths large at the end, turned with a medium coarse feed, the piston bored the proper size and shrunk on. With this method the ridges on the steel rod sink into the warm cast iron and make absolute contact, impossible to separate in ordinary service. The rod was then riveted over. I have used this practice on pistons up to 44 in. in diameter and never had a failure.

Some years ago a piston failed on a comparatively new engine. A crack being discovered a new piston was furnished by the builders. When received the piston was entirely finished except the bore, which was left below sizes for fitting. Feeling perfectly confident of a successful outcome, the defective piston was broken off, the rod trued as stated above and the piston bored to a shrink fit. The work was done by a very able workman, but I took the precaution to caliper the fit and decided the bore was too large. Another workman was called and he, like number one, decided the fit was all right. The piston was shrunk on and pronounced a good job.

They were then told to put the piston in the hydraulic press and see what pressure could be applied without starting the rod. Here, again, I was held up, as the jaw of the press was not large enough for the piston rod. I ordered the jaw chipped out, and started the rod with about six tons pressure. As the piston was nearly 30 in. in diameter a wrecked engine would have been the result. The engine had to be ready for service the following morning; and as no material was to be had for a new rod, and with the defective piston broken up, I was in a very uncomfortable position.

However, a thin steel liner was rolled to exact size of the piston bore, the piston again fitted and shrunk on the rod over the liner in the usual way, and again placed in the hydraulic press. Thirty tons failed to start the rod this time. The rod was riveted over and the engine put in service, and so far as I know, is still running and giving no trouble.

Los Angeles, Calif. H. P. CLARKE.

*"Power," Nov. 18, pp. 735 and 725.

Inquiries of General Interest

Heat Value of Natural Gas—What is the heat value of natural gas per 1000 cu.ft.?

V. K. S.

The calorific value of natural gas varies in different localities from about 1,100,000 B.t.u. to about 842,000 B.t.u. per 1000 cu.ft.

Height for Glass Gage on a Return-Tubular Boiler—At what height should the glass water-gage be set on a horizontal return-tubular boiler?

W. H.

So the lowest visible part of the glass will be at least 2 in. higher than the top of any tube of the boiler.

Pulsometer Trouble—Why should a pulsometer pump refuse to lift water excepting when the pump is submerged?

G. M.

The refusal of the pulsometer to operate unless submerged is probably due to leakage of the valves. The pump should be cleared of obstructions and the valves should be inspected and made tight.

Auxiliary Heating Surface—What is the usual method of adding pipe coil heating surface to a return-tubular boiler?

J. L. G.

Such auxiliary heating surface is usually added by placing pipe coils along the sides of the furnace walls and connecting them to both the front and rear ends of the boiler to insure more active circulation.

Rating of Feed Water Heaters—How many square feet of heating surface are allowed per horsepower for closed feed-water heaters?

M. S.

For commercial purposes, closed exhaust steam feed-water heaters are usually given a rating which allows about ⅓ sq.ft. of heating surface per horsepower.

Equivalent Evaporation—What evaporation from and at 212 deg. F. is equivalent to the usual boiler horsepower consisting of the evaporation per hour of 30 lb. of water from a feed-water temperature of 100 deg. F. into steam at 70 lb. gage pressure?

P. R.

Evaporation of about 34.48 lb. of water per hr. from and at 212 deg. F. would be equivalent to one boiler horsepower, but for ordinary calculations an evaporation of 34½ lb. of water from and at 212 deg. F. per hr. is usually taken as the equivalent.

Blowing Off Boiler Direct Into Sewer—What are the objections to connecting steam drips into a sewer or to blowing off a boiler directly into a pipe sewer?

J. G.

Most kinds of earthenware sewer pipes are rapidly disintegrated by the action of steam and hot water, and when a sewer pipe is connected with plumbing fixtures by iron piping, the latter is rapidly rusted away by the presence of steam and the back pressure due to blowing off causes disagreeable and unhealthy sewage vapors to arise from the plumbing fixtures.

Steel Steam Fittings—What is the difference between cast steel and the material of which steel steam fittings are made?

W. F. M.

The cast steel of fine grain such as was formerly used for the best cutlery, etc., is made by adding carbon to molten iron, after which treatment, the material is cast into billets and worked up into form by rolling and forging. Steel steam fittings are usually made by melting bessemer, openhearth or other kinds of steel and casting the molten metal in molds the same as in the manufacture of ordinary iron castings from molten cast iron.

Velocity of Entering Air—What is the maximum velocity at which air from a heating or ventilating system should enter a room? What are the reasons for such limitations of velocity?

M. N.

The velocity of air entering a room should not exceed 4 to 6 ft. per sec. unless the air can be so introduced as to make an insensible current. This is for the reason that higher velocities produce disagreeable currents, generally known as drafts, more or less dangerous to health and less efficient for purposes of heating or ventilating.

Wear of Steam Hose—What is the reason that for some uses a steam hose out of the same lot as another will develop leaks more rapidly when used for the same pressure and temperature of steam, and what is the remedy?

B. F. G.

The less durable hose is undoubtedly subjected to greater mechanical abuse, the commonest being to draw the hose out from a circular coil, causing twists and kinks. Twisting will not occur in drawing out the hose from a reel or when it has been hung up on two pegs by reeving the hose over the pegs in form of the figure 8 laid on its side.

Loss by Radiation from Covered Pipe—What loss of heat would occur by radiation from a 3-in. steam pipe 500 ft. long filled with steam at 338 deg. F., surrounded by atmospheric air at 80 deg. F. and covered with ordinarily good nonconducting covering?

G. D.

The bare pipe would have a surface of about

$$500 \div 1.09 = \text{about } 458 \text{ sq.ft.}$$

and the difference between the temperatures inside the pipe and of the external air would be

$$338 - 80 = 258 \text{ deg. F.}$$

The loss by radiation would be about 0.5 B.t.u. per sq.ft. of pipe per hour per degree difference, so that the total loss by radiation would be approximately

$$458 \times 258 \times 0.5 = 59{,}082 \text{ B.t.u. per hr.}$$

Factor of Evaporation—Having feed-water temperature at 50 deg. F. and evaporation at 100 lb. pressure per sq.in., absolute, what would be the factor of evaporation?

A. J. D. F.

From the "Marks and Davis Steam Tables," it is to be seen that to raise a pound of water from 32 deg. F. into steam at 100 lb. per sq.in., absolute, requires 1186.3 heat units (B.t.u.). With feed water at 50 deg. F., each pound of water raised into steam at a pressure of 100 lb., absolute, would receive 1186.3 — (50-32) = 1186.3 — 18 = 1168.3 B.t.u. To evaporate one pound of water from 212 deg. F. into steam at 212—i.e., at atmospheric pressure, or as commonly stated, from at 212 deg. F., requires for latent heat of evaporation 970.4 heat units. Hence, with feed water at 50 deg. F. and evaporation at 100 lb. pressure absolute, the factor of evaporation is

$$\frac{1168.3}{970.4} = 1.2039$$

Head Required for Water Pipe Line—What difference of head would be required to discharge 600 gal. of water per min. from one tank to another through a 7-in. galvanized iron pipe 2500 ft. long?

O. S. C.

The velocity would be practically 5 ft. per sec. Referring to tables given on page 55, July 9, 1912 issue, of "Power," the loss of head per hundred feet of 7-in. iron pipe due to discharging water at the rate of 5 ft. per sec. may be found to be about 1.47 ft., and for 2500 ft. of pipe it would be

$$\frac{1.47 \times 2500}{100} = 36.75 \text{ ft. loss of head.}$$

Flowing at the rate of 5 ft. per sec. would require a velocity head of about 0.39 ft., and assuming, as usual, that the loss of head due to friction at the entrance would be equal to one-half the velocity head—i.e., ½ of 0.39 = 0.19 ft.—then it is seen that the total difference of head required would be

$$36.75 + 0.39 + 0.19 = 37.33 \text{ ft.,}$$

or 37 ft. 4 in.

Engineers' Study Course

Trigonometry--VIII

LAST LESSON'S ANSWERS

31. In the solution of all right triangles it is a help to keep in mind the typical triangle (Fig. 2, Lesson I; reproduced herewith as Fig. 7). The lettering can be easily remembered by noticing that the side b is the *b*ase and the side a the *a*ltitude. Then c is the only side left (the hypotenuse) and the angles have the same letters (but capitals) as the sides opposite them. The functions can then be remembered by the shorter forms given in

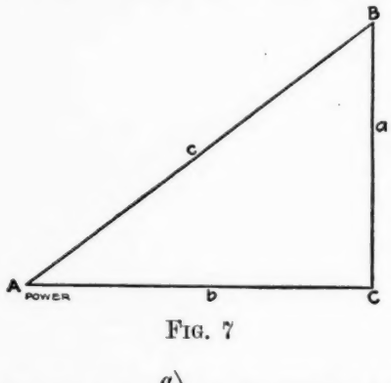


FIG. 7

Lesson I (as $sin\ A = \frac{a}{c}$) more readily than by those given in the last lesson (as $sine = \frac{opposite\ side}{hypotenuse}$).

Given $sin\ A = \frac{1}{2}$ and $c = 36$ in. Since $sin\ A = \frac{a}{c}$.

$$\frac{1}{2} = \frac{a}{36\ in.} \text{ or } a = 18\ in.$$

and the third side

$$b = \sqrt{c^2 - a^2} = \sqrt{36^2 - 18^2} = 31.177\ in.$$

The sine of the other oblique angle will be its opposite side divided by the hypotenuse, or the same as the cosine of A, or

$$sin\ B = cos\ A = \frac{b}{c} = \frac{31.177}{36} = 0.866$$

As a check, refer to the table in Lesson II. It will be found that the acute angle whose sine is $\frac{1}{2}$ is 30 deg. Therefore, the other acute angle must be 90 — 30 = 60 deg. and, from the table, *sin* 60 deg. = 0.866.

32. Given the base ($b = 20$) and the altitude ($a = 32$). The hypotenuse (c) will be

$$c = \sqrt{a^2 + b^2} = \sqrt{32^2 + 20^2} = 37.736$$

$$sin\ A = \frac{a}{c} = \frac{32}{37.736} = 0.848$$

From the table (Lesson II) the angle corresponding to this sine is 58 deg. The other acute angle is therefore

$$90 - 58 = 32\ deg.$$

33. Given $c = 235$ and $A = 43$ deg. The other acute angle will be

$$B = 90 - 43 = 47\ deg.$$

From the table $sin\ A = 0.682$. Then, since $sin\ A = \frac{a}{c}$,

$$0.682 = \frac{a}{235}, \quad a = 0.682 \times 235 = 160.27$$

From the table *sin* B (or *cos* A) = 0.7314. Then, since $cos\ A = \frac{b}{c}$,

$$0.7314 = \frac{b}{235}, \quad b = 0.7314 \times 235 = 171.879$$

If it is desired to check the work, it can be seen if the sides agree in the relation

$$c^2 = a^2 + b^2$$

The method used in the solution of this problem suggests two rules for similar ones:

Given the hypotenuse and one angle of a right triangle, to find the opposite side, multiply the sine of the angle by the hypotenuse, and, to find the adjacent side, multiply the cosine of the angle by the hypotenuse.

34. Given $A = 47$ deg., and $b = 180$ ft. From the table *cos* 47 deg. = 0.682. Since $cos\ A = \frac{b}{c}$,

$$0.682 = \frac{180}{c}, \quad c = \frac{180}{0.682} = 263.93\ ft.$$

From the table *sin* 47 deg = 0.7314. Then, applying the rule just given,

$$a = sin\ A \times c = 0.7314 \times 263.93 = 193.04\ ft. = height\ of\ chimney$$

35. In effect the problem states that the connecting-rod (the hypotenuse c) is 1½ in. longer than the distance from the center of the crosshead pin to the center of the crankshaft (the base b).

Given, then, $a = 8$ in. and $b = c - 1\frac{1}{2}$ in. Also $b = \sqrt{c^2 - a^2} = \sqrt{c^2 - 64}$.

Then

$$c - 1\tfrac{1}{2} = \sqrt{c^2 - 64}$$

Squaring both sides

$$c^2 - 3c + \tfrac{9}{4} = c^2 - 64$$

Canceling c^2 and transposing

$$64 + \tfrac{9}{4} = 3c \quad 3c = \tfrac{265}{4} \quad c = \tfrac{265}{12} = 22.08\ in.$$

Required angle A. Since $sin\ A = \frac{a}{c}$,

$$sin\ A = \frac{8}{22.08} = 0.3623$$

From the table, A is between 21 and 22 deg., for *sin* 21 deg. = 0.3584 and *sin* 22 deg. = 0.3746.

The total difference between the sines of these two angles is

$$0.3746 - 0.3584 = 0.0162$$

The difference between the smaller and the calculated *sin* A is

$$0.3623 - 0.3584 = 0.0039$$

Therefore, the angle A exceeds 21 deg. by about $\frac{0.0039}{0.0162}$', or approximately ¼ deg. A reference to a table of natural functions with greater subdivision of angles would show that the angle A is very closely 21¼ deg., or 21 deg. 15 min., nearly.

Solution of Oblique Triangles

As distinguished from right triangles, all other triangles are known as *oblique triangles.*

It was stated in Lesson I that "if any three of the six parts (the three angles and the three sides) of a triangle are known, and at least one of them is a side, the shape and size of the triangle are determined." If one were given three such parts one could construct the triangle and measure the unknown parts with a protractor for the angles and a scale for the sides. It remains to be shown how the unknown parts can be calculated from the known without any direct measurements.

As for the angles, when two are known, the third can be found at once by subtracting the sum of the two known angles from 180 deg., for the three angles of any triangle total 180 deg. Each angle of a triangle is therefore the *supplement* of the sum of the other two, as explained in Lesson II. That this is true of right triangles was shown in the answer to question 10 in Lesson III. To prove that it still holds for any triangle, refer to Fig. 8. $ACDA$ and $BCDB$ being right triangles the sum of the angles of each totals 180 deg. and the two together 360 deg. Subtracting the two right angles ADC and CDB, which do not figure in the triangle $ACBA$, and there is left

$$360 - 90 - 90 = 180 \text{ deg.}$$

Another proof:

Angle ACD being the complement of angle CAD and DCB the complement of CBD

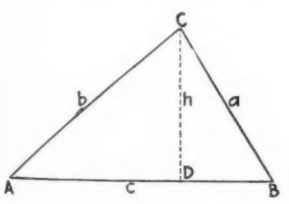


Fig. 8

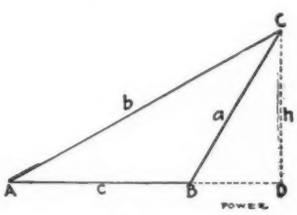


Fig. 9

$$\angle ACD + \angle CAD = 90 \text{ deg.}$$
$$\angle DCB + \angle CBD = 90 \text{ deg.}$$

Therefore,

$$\angle ACD + \angle CAD + \angle DCB + \angle CBD = 180 \text{ deg.}$$

but

$$\angle ACD + \angle DCB = \angle ACB$$

therefore,

$$\angle CAD + \angle ACB + \angle CBD = 180 \text{ deg.}$$

Either of these proofs would apply to the triangle $ACBA$ in Fig. 9 if a perpendicular were drawn from the angle B to the side b, but the proof may also be made from the figure directly as it is as follows:

$$\angle ACD + \angle CAD = 90 \text{ deg.}$$
$$\angle BCD + \angle CBD = 90 \text{ deg.}$$

Therefore,

$$\angle ACD + \angle CAD = \angle BCD + \angle CBD$$

Subtracting $\angle BCD$ from both sides of the equation

$$\angle ACD - \angle BCD + \angle CAD = \angle CBD$$

but

$$\angle ACD - \angle BCD = \angle ACB$$

Therefore,

$$\angle ACB + \angle CAD = \angle CBD$$

Then, since

$$\angle ABC + \angle CBD = 180 \text{ deg.}$$
$$\angle ABC + \angle ACB + \angle CAD = 180 \text{ deg.}$$

For the complete solution of any oblique triangle there are four cases which, according to the three known parts, are as follows:

Case I. When one side and two angles are known.

Case II. When two sides and the angle opposite to one of them are known.

Case III. When two sides and the included angle (the angle between them) are known.

Case IV. When the three sides are known.

Cases I and II

Solutions of triangles having one side and two angles given, or two sides and the angle opposite one of them given, require the use of what is known as *The Law of Sines,* which is stated thus:

The sides of a triangle are proportional to the sines of the opposite angles.

This may be proved by referring to either Fig. 8 or Fig. 9, in both of which $ABCA$ represents any triangle, from any angle, as C, of which a perpendicular h is drawn to the opposite side, or that side extended.

From the right triangle $ACDA$, $h = b \sin A$, and from the right triangle $BCDB$, $h = a \sin B$.

(In Fig. 9 $\sin ABC = \sin CBD$).

Equating the values of h

$$b \sin A = a \sin B$$

which, by transposition, becomes

$$\frac{a}{b} = \frac{\sin A}{\sin B} \qquad (27)$$

In the same way it could be shown that

$$\frac{a}{c} = \frac{\sin A}{\sin C} \text{ and } \frac{b}{c} = \frac{\sin B}{\sin C}$$

Regarding these three equations as proportions

$$\frac{a}{\sin A} = \frac{b}{\sin B} = \frac{c}{\sin C}$$

Study Questions

36. Referring to a triangle lettered as in Fig. 8, given $a = 500$, $A = 10$ deg. and $B = 46$ deg., find C, b and c.

37. Given $b = 137.5$ ft., $B = 13$ deg. and $c = 512.5$ ft., determine the other three parts of the triangle.

38. In a triangle lettered as in Fig. 9, given $c = 23$, $A = 23$ deg. and $B = 134$ deg., find a, b and C.

39. To determine the distance across a river between two transmission poles, a distance of 100 ft. along one bank was measured from the base of the near pole and the angles which sights on the distant pole were found to make with the extremities of this line proved to be 45 deg. at the end of the line and 85 deg., at the near pole. What was the distance between the poles?

40. An engine comes to rest beyond the center of a forward stroke. The crank, which is 18 in. long, is at an angle of 45 deg. with the horizontal. The connecting-rod is at what angle with the horizontal, if 45 in. long?

OVER THE SPILLWAY

JUST JESTS, JABS, JOSHES AND JUMBLES

ENGINEER WANTED, small plant; one who is willing to do porter's work. Apply 66 University place.

And then they prate of the dignity of labor! This is a bona-fide ad from the New York "World." Now, don't throw up your steady job unless you are sure you can cop this one. The line forms on the right.

⁂

Every now and then a highly peeved party complains that he has asked us to answer an operating inquiry, and he gets highly indignant because we ignore him. Suppose you received a letter signed "A. S. S.," no town name, no street or number, no nothing nohow. There's more than one "A. S. S." in these United States (get me?), so whom would you write to? One week's subscription to the best answer to this inquiry.

⁂

Walter H. Damon, over to Springfield, has sprung into the arena announcing his candidacy for the vice-presidency of the N. A. S. E. There's some spring to Walt, so you other candidates will have to get busy if you hope to make a showing.

⁂

Here's the chance of their lives for the jokesmiths: The Yuba River has been dammed! Yuba dam! just like that. Nothing ever escapes them, and you may as well be prepared to meet up with this information whichever way you turn.

⁂

The editor of a mining journal has taken his pen in hand to cast contumely on the Finn, who "overbalances his undeniable intelligence and skill with an unextinguishable desire to start something." We see this editor's finish, if ever an irate Finn fastens his fin on him; some Finns are real finnicky.

⁂

It is said that the inventor of the cash register got his inspiration "through close observation of the distance-recording device in the boiler room of an ocean liner." But the other fellows got the fortune, which is the way of most inventors. Whadcha mean, genius is not practical?

⁂

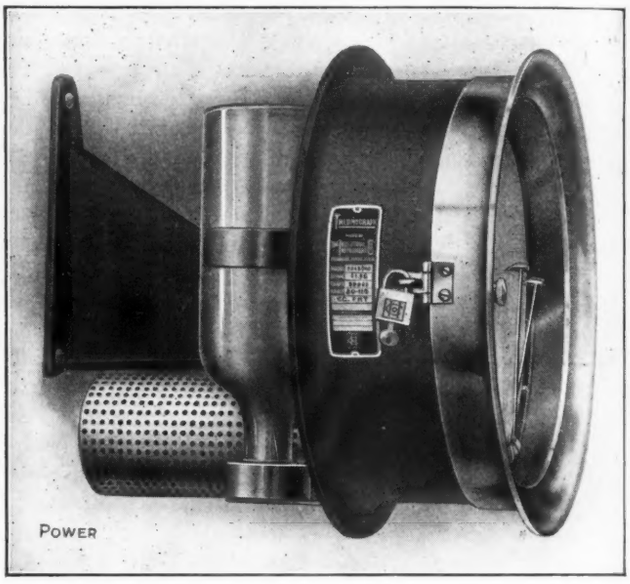
WE'RE DOING BETTER EVERY MONTH JOHN. KEEP THE GOOD WORK UP.
JOHN! DON'T THINK YOU'RE DOWN TO THE PLANT SHOVELING COAL. LOOK AT THIS BILL!
COAL BILL
POWER

"THE FUEL SITUATION"

Improved Recording Hygrometer

The Foxboro hygrometer is, in reality, two instruments in one, a recording thermometer and a wet-bulb recorder. The illustration shows the standard form of stationary, self-contained type. It consists of two sensitive bulbs mounted in tandem back of the case, the wet bulb being jacketed and kept moist by maintaining water at a constant level in a trough beneath. The pen arms are attached to shafts concentric with the helical tube bulbs.

The case is mounted on a swivel bracket, enabling swinging the instrument at right angles to the wall or



IMPROVED RECORDING HYGROMETER

support, giving easy access to the inverted glass bottle serving as the water reservoir. It is made in three sizes, 8, 10 and 12 in., and covers ranges between the freezing point and the boiling point of water. The instrument is made to ascertain the relative humidity in textile manufacturing, etc., by the Industrial Instrument Co., Foxboro, Mass.

⁂

Elusive Knocks in Ammonia Compressors

BY THOMAS G. THURSTON

Coming into the power house one morning the night engineer informed the writer that the spring on one of the discharge valves of one of the ammonia compressors was broken and that it would be necessary to get a new one. The compressor was a horizontal double-acting machine of 250 tons capacity and the noise it was making gave every indication of a valve slamming on its seat when the spring is broken.

The machine was pumped out and when the bonnet was removed we were surprised to find that the spring was all right. However, to make sure a new spring was put in place. Upon starting up again the noise was as bad as ever and for a while we were at a loss to locate the trouble. Finally, the oiler accidentally solved the problem by drawing up on the stuffing-box nuts to stop a leak at this point. The machine had been in service for several years and the compressor piston rod had been turned

down a number of times, so that it was considerably smaller than the hole through the cylinder head. To keep the packing from going through into the cylinder a junk ring had been fitted into the bottom of the stuffing-box and made a snug fit on the rod. It was this junk ring that had been making all the noise. The packing working loose had allowed the ring to move and at the reversal on the crank-end stroke it would strike the bottom of the stuffing-box with a slam, identical to that of a discharge valve with a broken spring.

Coming on duty one morning a couple of weeks later the night engineer had another case of a broken discharge-valve spring. This time it was on a 200-ton vertical machine, and, according to the night engineer, it was one of the top discharge valves. There was a sharp knock and upon standing on the platform beside the cylinder it seemed to be in every place in which we tried to locate it. The conclusion was finally reached that it was in the lower suction valve and the machine was pumped out and the valve cage removed. The valve was all right, but again to make sure a new one was put in place. The knock was still evident when the machine was started up again so the same performance was repeated on the top suction valve with the same result. The bottom discharge valve was tried and when the machine was started the knock was worse than ever.

Finally the top head was removed and the top discharge valves examined. One of them showed a tendency to stick and naturally we thought that the trouble had been located. Upon starting up the knock was as loud as ever. It came after the crank was well past the center and was surely some place in the cylinder, but where was the question. Several brilliant suggestions were advanced by the members of the engine-room force and a number of methods tried to locate the knock without success. I finally went below the cylinder and here the knock was more pronounced and seemed to come from the stuffing-box where it could be felt distinctly. The machine was again pumped out and the first half of the packing removed. The oil lantern stuck to the rod, but we finally got it down. At some time the rod had become too hot and the babbitt in the lantern had softened and when it cooled it had wedged on the rod. When the lantern had been lowered the cause of the trouble was evident. There was no packing back of the lantern and as it stuck to the rod it banged against the bottom of the box at every upstroke of the piston rod. The rod was packed in short order and the trouble was over.

Inspecting a Return-Tubular Boiler, Part I

By James F. Hobart

SYNOPSIS—Information as to how an inspector's suit should be made, what tools are necessary, details to observe and some of the defects expected to be found.

Steam-boiler inspection is for the purpose of detecting defects which might lead to possible accident. There are two classes of boiler inspection, that when the structural details are known, and that of unknown boilers. Before an intelligent inspection of an unknown boiler can be made a data sheet must be made out from which calculations of the strength of riveted seams may be determined, also the strength of the braces and stays, and the unsupported areas which are not measured and listed.

When the structural details are known, the boiler should be examined thoroughly, both inside and out, if the shell is large enough to admit of internal inspection. If it is a small vertical boiler then an external inspection and an examination through the several openings in the boiler shell must answer.

First of all a boiler shell should be thoroughly cleaned on the outside by sweeping off all accumulated soot and ashes. It is impossible to properly inspect a boiler when coated with soot and ashes, or when crawling around knee deep in combustion-chamber ashes.

For internal inspection a return-tubular boiler should first be thoroughly washed out and allowed to become reasonably cool by removing the handhole and manhole plates, opening the doors in front of the tubes, and adjusting the dampers and furnace doors so that cold air will be drawn into and through the boiler as well as around it. This will do no harm after the shell has cooled down to 212 deg.

Inspector's Kit

A complete dust-proof suit will help make the task of inspecting a boiler more agreeable. It usually consists of a union suit of overalls and jacket, made to fit closely around the throat, wrist and ankles. A cap fits tightly around the face so that dust and dirt will not work into the hair. Fig. 1 will give an idea of the manner in which an inspector's hood is made.

FIG. 1—CAPE HOOD

FIG. 2—WRIST FASTENING

FIG. 3—METHOD OF MAKING FRONT OF SUIT

FIG. 4 ANKLE FASTENING

POWER

Details of Inspection Suit

A cape of ample proportions should be attached to the hood and the caps should button in front and fit closely over the shoulders of the wearer. Fig. 2 shows the manner of fitting the suit around the wrist. Some are secured with a strap, others by a cord. The suit should be buttoned in front from top to bottom, and cut with a lap of several inches, as shown by Fig. 3, to prevent dirt from penetrating the suit.

The legs of the overalls are made long enough to reach to the floor and drag at least an inch when standing erect. Then the cloth is fastened closely around the ankles of the wearer. Two methods are used for fastenings at the ankles. One consists of tying a common shoe string around the ankle. The other method consists of a three-strap arrangement, two being fastened to two rings. One strap passes behind the ankle, and the other underneath

FIG. 5
STEAM GAGE JACK

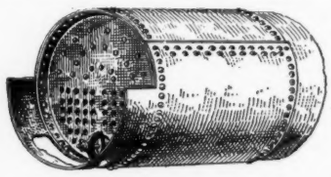


FIG. 6—BURNED OUT DRY SHEET

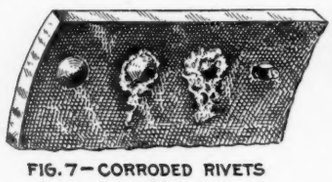


FIG. 7—CORRODED RIVETS

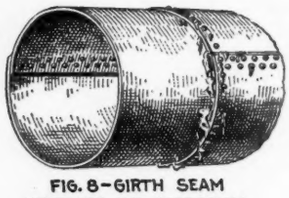


FIG. 8—GIRTH SEAM
DISTRESS AND CORROSION

POWER

DEFECTS TO GUARD AGAINST

the foot just forward of the heel. When the third is buckled closely it draws the fabric around the ankle and also prevents the leg of the overalls from pulling upward while crawling through restricted passages.

Another arrangement is that of using two straps, Fig. 4. The front strap is omitted and replaced by a cord or a piece of leather-belt lacing which is passed through the rings, crossed over the instep, wound around the ankle, and then tied in front. This fastening is almost dirt-proof. High shoes should be worn, and the overalls drawn down outside and fastened.

Certain tools are required for inspection work, as, for instance, a peen hammer weighing about 1 lb. The peen should be crosswise of the hammer, and an ordinary riveting hammer answers very well. A small screwdriver is also necessary to use on the screws of steam gages; a small wrench or a pair of gas pliers can be used to disconnect steam gages.

Another tool is a little jack, Fig. 5, for removing the hand from a steam gage without bending the pinion, which is almost sure to be done if the attempt be made to remove the hand with a pair of pincers, knife or a claw hammer.

A gage-testing outfit is also necessary for testing steam gages. A light but strong steel chain about 6 ft. long, and a padlock should be used to lock the boiler stop valve, when there is more than one boiler on the line, so that the valve cannot be opened. A similar precaution may be taken with the feed-water pipe, but usually the removal of the check-valve cap offers ample protection against hot water entering the boiler.

WHAT TO LOOK FOR

The first thing to note is the condition of the boiler setting. The condition of the brickwork should be examined, also the fire surface of the shell, for signs of blistering, bulging or of burnt or corroded rivets. Specially should the front of the furnace be looked at, for, if the brickwork burns out or falls down, the dry sheet will burn, as shown in Fig. 6, permitting flames to pass directly from the furnace to the front smoke arch. This cannot happen with a full flush-front boiler.

Next the seams should be inspected for signs of distress. Overheating seams frequently causes leaky rivets, and, although the rivets may have tightened up again, corrosive deposits may be found frequently around them. In locomotive and internally fired shells, very close search should be made for bulged plates, distorted sheets, signs of burning and bagging of plates between the stay-bolts or braces.

The cause of such bulging should also be closely looked for. This will probably be found during the internal inspection and is usually due to dirt or scale in the boiler adhering to the plate and preventing water from reaching the shell. When that happens, undue heating takes place and the boiler shell may heat sufficiently to permit the internal pressure to bulge out the plate. If a bag is detected it should be examined closely and noted if, when struck with a hammer, it gives a sound indicating cracks or thin places. Small bulges are usually left alone; large ones must be removed because the pocket formed in the sheet collects sediment, which causes further overheating and bulging. Cutting out a bulge may not prove very satisfactory, as a pocket is still left which may cause trouble indefinitely. Forging a bulge back is easily done if the sheet is heated to a bright red.

CORROSION

All longitudinal seams of the boiler should be looked after to ascertain what corrosion, if any, exists. Along each of the outside straps there may be signs of rusting of the metal or a deposit of white-looking matter. Sometimes the deposit will be yellow or dark colored, depending upon the corrosive substance which has been acting upon the shell.

The peen of the hammer should be used to cut away the deposit and to determine how much, if any, the sheet has been reduced by the corrosive action. A steel scale is handy for this purpose. It can be placed edgewise against the boiler shell, one end of the scale extending over the corroded spot, then the distance from the scale to the boiler plate will equal the amount which has been removed by corrosion.

Corrosion of the boiler shell should be watched for closely. This is to be found either where the plate laps over on itself or where one comes against the inner or outer strap. By placing a corner of the steel scale against the joint, and parallel with the boiler it is easy to judge how much or how little wasting of plate has taken place. This is a very important part of the work, as a reduction of $\frac{1}{16}$ in. in the thickness of the boiler plate may cut down the allowable pressure enough to require taking the boiler out of commission.

The examination shows the physical condition of the boiler, and, from the data acquired, the safe working pressure for the boiler is calculated. The pressure carried in the boiler should be cut down to the amount thus determined, as a boiler is no stronger than the weakest spot.

CORRODED RIVETS

In Fig. 7 are shown corroded rivets, also one in good condition. Sometimes one is found as indicated by the second, almost completely covered with a white powdery-looking deposit which may go partly or completely around the rivet. Frequently greater action takes place, and the

appearance shown by the last rivet is often found, and the top of the rivet can barely be seen in the center of a mass of corrosion.

When such rivets are found they should be removed, the holes reamed slightly and new rivets driven. Repairs should be made to all defective rivets. Sometimes the metal will be eaten away on two sides of the holes, as shown. It is possible to drive a new rivet in such a hole and very often the rivet fills the hole so tightly that no leak will take place, but it is much better and safer to ream the hole slightly, until it becomes round again, then drive a new and larger rivet.

Each girth seam should be looked at closely, not only for corrosion but for signs of distress.

Inspecting Girth Seams

Fig. 8 shows some things which one may expect to find about the girth seams. The deposits are caused by evaporation of the water as it issues from leaks in the boiler, leaving deposits of salts which fill the angle between the sheets or around the rivet heads. When leaks exist along a seam or beside a rivet, corrosion must eventually take place. Water cannot be forced through a crack in a hot metal plate without uniting with the metal of the plate to a greater or less extent. Some of the rivets may also show deposits. Such conditions indicate distress at the girth seam, and the cause should be closely investigated.

Usually the girth seams of boilers give little trouble as all they do is to hold the boiler together and withstand the pressure on the heads not carried by the tubes. Therefore, the inspector should look for other reasons for the distress shown at girth seams, such as low water at some previous time, for overheating will start girth-seam and rivet leaks very quickly. The method in which the boiler is suspended figures in the trouble. Sometimes boilers are so placed that expansion of the shell is restricted and excessive strains are caused in the girth seams.

Powdered Coal in Boiler Furnaces

Powdered coal, with suitable apparatus, can be made to produce complete combustion in boiler furnaces. This is no new discovery. About twenty years ago the practice began to develop and it is about fourteen years since C. Wegener introduced a powdered-coal-burning system in Europe wherein the use of blowers for forcing the coal to the furnace was eliminated, and the natural draft made to do this function. Then followed an interesting development of apparatus for using this form of fuel, and this activity still flourishes despite the little success in the practical application of the art. At present much experimenting is being carried on.

This manner of burning coal has many features to recommend it. Not only will coal burn better and more completely when pulverized, but coal high in volatile gases, when ground, will give up to the boiler much of the heat in these gases that goes to waste under ordinary firing and draft conditions. The draft must be very much lower when powdered coal is burned, for the volatile combustible burns more rapidly than the fixed carbon in powdered form, which will be carried unconsumed to the back of the setting even under the low draft. The coal must not enter the furnace at too high velocity and, to prevent rapid deterioration of the firebrick, must not impinge on any part of the setting. The ideal condition is one where the fuel burns completely in suspension.

The economical utilization of the cheaper grades of bituminous and slack coals is another important reason why a successful powdered-coal-burning apparatus would be welcome. True, these cheap fuels are being more extensively used, especially with the more general use of the stoker, but the high percentage of ash and moisture are productive of trouble. When in powdered form and suitably handled, the high ash and moisture content are not seriously objectionable except that pulverizing troubles arise from too high moisture content. The temperature in parts of the furnace when using powdered coal is often much higher than when burning ordinary fuel with stokers or by hand firing. As the fusing temperature of the firebrick is sometimes lowered by the combined action of the chemical constituents of both ash and firebrick, the composition and the fusing temperature of the ash are important.

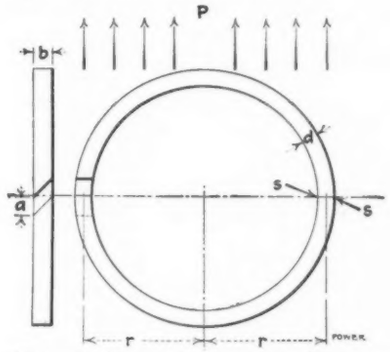
With coal high in ash that slags, it becomes important to run off while hot the slag that accumulates in the bottom of the setting, especially if the boiler must be kept in continuous service for long periods. Lime, unslacked or as contained in clam and oyster shells, used as a flux will usually soften the slag so that it will flow or become plastic enough to permit of being pulled out with a hoe.

The evaporation per pound of powdered coal rightly burned is high, the labor cost is comparatively low and there is almost a total absence of smoke when burning the highly volatile coals in powdered form. While the years of endeavor have not resulted in the general adoption of this method of burning coal in boiler furnaces, the persistency with which experiments are carried on may result in the perfection of the art, so that it will prove superior to the stoker for burning many of the lower grades of fuel that now lie in mountainous piles in the coal regions.

Proportioning Piston Rings

To calculate the pressure of piston rings against the cylinder walls, let

P = Total pressure of ring, in pounds;
p = Unit pressure of ring, pounds per square inch;
S = Unit stress at most remote fiber, pounds per square inch, at point s;



Piston Ring, Showing Where Measurements Should Be Taken

c = Distance from neutral axis of section to most remote fiber, inches;
I = Moment of inertia of section about neutral axis parallel to the face;
M = Maximum bending moment;

r = Mean radius of ring, inches;
a = Amount to be cut out of ring;
D = Diameter of cylinder bore;
b = Face of ring, inches;
d = Thickness of ring, inches;
F = Allowance for turning.

The fundamental flexure formula for beams is $S = \frac{Mc}{I}$.

Consider one-half of the ring as a cantilever beam of length $2r$, and uniformly loaded with a load, P.

M for cantilever beams $= \left(\frac{P}{2}\right) \times (2r) = Pr.$

c for rectangular sections $= \frac{d}{2}$.

I for rectangular sections $= \frac{(bd^3)}{12}$.

Substituting in $S = \frac{Mc}{I}$.

$$S = \frac{\frac{Prd}{2}}{\frac{(bd^3)}{12}} = \frac{\frac{Pr}{2}}{\frac{(bd^2)}{12}}$$

Reducing P to pounds per square inch projected area $= 2rbp$. Substituting

$$S = \frac{\frac{(2rbp)\,r}{2}}{\frac{(bd^2)}{12}} = \frac{pr^2}{\frac{d^2}{12}}$$

Solving for p,

$$p = \frac{Sd^2}{12\,r^2}$$

The amount a to be cut out of the ring may be assumed some even figure from about 3/8 in. for the smaller sizes of cylinders to 3/4 or 7/8 in. for the larger sizes of 16- or 20-in. bore.

After assuming the value for a, the outside diameter of the cylindrical blank from which the rings are cut would be

$$D' = D + \frac{a}{3.1416} + F$$

The rings are then cut off with the required width b, and, after the slot a is cut, they are sprung together and turned to the same diameter as the cylinder bore.

Compounding Simple Reciprocating Engines

By L. H. Morrison

SYNOPSIS—How two engines of different capacity and running at different speeds were piped to run as a compound condensing unit. The directly connected generators were run in parallel without trouble, notwithstanding predictions made to the contrary.

About a year ago the writer assisted in designing a municipal electric-light plant, some features of which are worthy of mention.

A city of approximately 6000 inhabitants purchased the franchise and physical property of the Electric Light & Water Co., and found it necessary to build an entirely new plant. It was also determined to install a central steam-heating system to operate in conjunction with the lighting plant, this system to cover all the business district and a part of the residential section.

It was desirable that the generating units be the most economical obtainable with the amount of money available. Furthermore, to determine the capacity of the new units, recourse was had to the old company's records which revealed that during the winter the maximum demand was about 350 kw., and the maximum required during the summer was approximately 225 kw. The average consumptions during these two periods had about the same ratio.

The sizes of the units necessary was the first thing to be considered, and a 100- and a 225-kw. unit were decided upon. The two units combined were ample to carry the maximum winter load with the prospective increase for several years. On the other hand, the 100-kw. set could carry the light day load in summer and the 225-kw. unit was ample to handle the heavy summer load. It was imperative that two units be installed as a safeguard against a total shutdown, with the additional reason that it was a good business proposition to have that unit in operation at any time that it would be carrying approximately full load.

As a first step toward choosing the type of prime mover to be installed, the turbine was investigated and discarded, because of the high steam consumption of the smaller sizes, especially when operated noncondensing and with a back pressure. Although the steam consumption was of small moment during the two months of extremely cold weather, during the autumn and spring the steam necessary for the heating system would be small; consequently the economy with noncondensing operation was of importance and more than offset the low installation cost of the turbines.

Similarly the high-speed, single-valve engine was discussed and discarded in view of the ever increasing steam consumption of this type, while even the consumption, when the engine was new, was entirely too high for this plant. This narrowed the field to two types, the Corliss and the four-valve engine.

Investigation seemed to prove that as regards cost of operation and upkeep there was little to choose between these two types. In view of the demand that the cost of current be as low as possible, the compound engine seemed almost essential. However, the money available was insufficient to purchase two compounds and, as has already been stated, it was desirable to have the two units. Finally it was decided to purchase two simple engines and arrange to operate them part of the time as a compound engine.

This idea was taken up with various engine and generator builders, but none would vouch for the success of

the arrangement. The engine people stated that the engines would operate satisfactorily if the generators did not develop heavy cross-currents. The generator people were quite willing to guarantee the operation of the alternators if the engines would regulate.

All stated that it was a proposition entirely new to them. However, in spite of the lack of builders' guarantees it was decided that the plan should be carried out. It was impossible to buy two compound engines and any gain in steam consumption with the compounding plan over simple engines would be clear gain. Furthermore, the cost of a receiver and the necessary piping was not excessive.

Bids were invited on the installation of two 72-in. by 20-ft. horizontal tubular boilers for 175 lb. working pressure, a 100- and a 225-kw. engine and generator unit together with condenser, piping and the usual accessories. The generators were to be 2300-volt, three-phase, 60-cycle, directly connected to the engines and to have belted exciters. The bidders were required to give steam consumptions with engines operating both noncondensing and condensing. The city assumed responsibility for the compounding arrangement and so this plan was not discussed with the bidders.

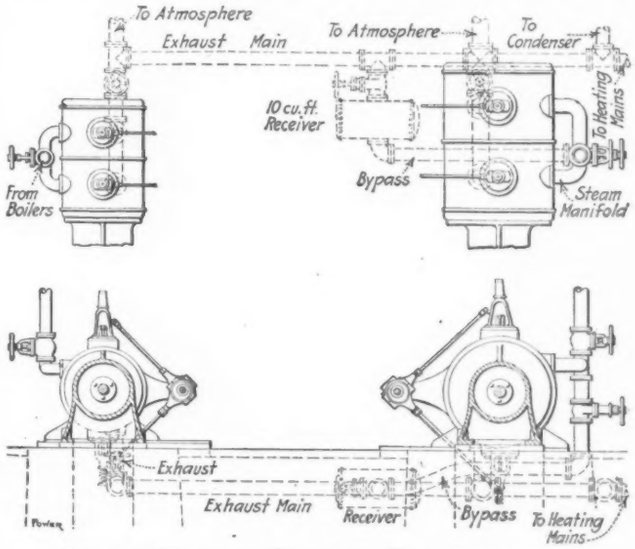
Finally, because of extremely low steam consumption and evident high-grade of workmanship entering into the construction, two improved, four-valve German-design engines were purchased with the other machinery called for. The engines were a 12x18-in., operating at 225 r.p.m., directly connected to a 100-kw. alternator, and a 19x27-in. engine directly connected to a 225-kw. alternator running at 150 r.p.m. It was desired to have both engines run at the same speed, but the cost of the smaller unit at 150 r.p.m. was so high as to over-run the available money.

Both engines were connected to a main exhaust line, as shown in the illustration. There was also an independent atmospheric exhaust line from each engine. The main exhaust line, immediately beyond the large unit, was connected to a jet condenser. The bypass with its connecting receiver is shown. This bypass was connected to the live-steam manifold of the large engine from below, this engine being equipped with two throttle valves, one controlling the line from the boilers, the other controlling the steam from the bypass.

The method of operating the plant is as follows: During the season of steam heating, the engines are operated noncondensing and exhaust directly into the heating mains At times one engine handles the load, but during the greater part of the time both are operated with the generators run in parallel. During the warm months when the load is less than 75 kw., the small unit is operated condensing. When the load increases, the two engines are started on live steam, and the generators are then thrown in parallel. This being accomplished, the small-engine exhaust is cut off from the large-engine exhaust beyond the bypass, the exhaust passing out through an independent atmospheric line through a back-pressure valve, the weight of which can be altered at will. The bypass is then opened and the live-steam throttle gradually closed. When the pressure in the bypass reaches about 30 lb. or more the live-steam throttle is entirely shut off, and the bypass throttle opened. The weight on the back-pressure valve of the small engine is then shifted so that it will not open with less than 80 lb. pressure.

As the back-pressure instantly shows up on the small engine or high-pressure cylinder, the cutoff occurs later, according to the load on the generator. The same occurs with the large engine or low-pressure cylinder. The capacity of the receiver, 10 cu.ft., prevents fluctuations of the receiver pressure that were expected to occur, due to the different engine speeds.

It had been freely predicted that the regulation would be faulty, especially that of the large engine. On the contrary, the two units operated as smoothly as when running as simple engines. It developed that the small engine's governor shifts often while the large engine governor's movement is very slight. This is probably explained by the fact that as the load decreases the cutoff



SIMPLE ENGINES PIPED TO RUN COMPOUND

of the small engine becomes shorter, thereby lowering the terminal and receiver pressure; this reduction of pressure would decrease the power of the large engine with the same cutoff as before. Consequently the governor is not shifted as frequently. The cross-currents between the generators do not develop as many had feared. In fact, the operation is as good as found with many regular compound engines.

While the steam consumption of the combined units is about 2 lb. higher than that of a compound engine, it is very evident that the greater part of the preventable loss is due to the lowered efficiencies of the two generators, since they are not operated at full load. Also, there would probably be a slight gain if the engine cylinders had a better ratio and the revolutions were the same.

It is now proposed to run the exhaust-steam line from the condenser pump into the receiver, thereby giving an additional amount of steam to the low-pressure cylinder; the exhaust from the boiler-feed pump being ample for heating the feed water.

While it is a debatable question if such an arrangement of compounding is often advisable, still in this case it works satisfactorily and indicates the flexibility that it is possible to secure in a medium-size light plant.

How to Take Rope From a Coil Without Kinking—The Plymouth Cordage Co. gives the following directions: Place the coil on its head with the inner end of the rope at the bottom, then draw out the end anticlockwise and throw the rope as it comes out so that it will fall in the same direction as that taken during the unwinding. Where storage space is limited it may be necessary to draw from the outer end of the rope.—"Mill Supplies."

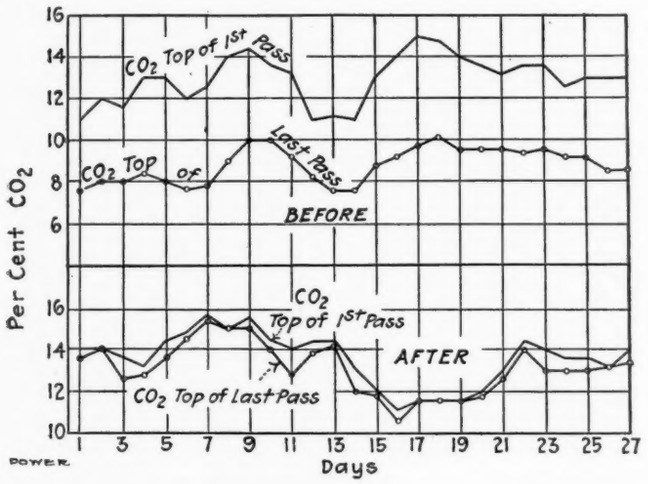
Air of Infiltration

By Morgan B. Smith

SYNOPSIS—Description of and data from tests showing effects of air leakage into boiler settings considered reasonably tight, and the method employed to secure practically absolute air-tightness.

In a plant of some 20 odd boilers aggregating 7000 hp. (rated), two boilers continually showed a lower efficiency of combustion than the rest, although all were similarly constructed and operating on the same fuel. The flue gases were analyzed daily for CO_2 with a recorder and frequent tests were made with the Orsat for CO_2, CO and O. Finally, two Orsats were employed, one for the gases at the top of the first pass, the other just before the back damper at the top of the last pass.

The tests at once showed that the gases as they left the boiler had become diluted as compared with the gases at



CO_2 Record for 27 Days Before and After Setting Was Made Tight

the point of gas travel where combustion was practically just completed. The dilution was caused by air, as shown by the analyses of the gases, and inasmuch as none had been admitted through the furnace which had not been accounted for in the analyses taken at the top of the first pass, it is evident that air which had not passed through or over the fire bed must have found its way into the boiler through the setting itself.

We at once began a thorough investigation of this setting and prepared to stop the entrance of any air except that which rightfully came in through or over the fuel bed. We found bad joints in the brickwork, and porosity tests of bricks similar to those used in the setting showed that considerable air could readily pass through them when under the influence of difference in pressure; door frames were loose in the brickwork, and we located numerous points of entrance for air which before had passed unnoticed.

The first step was to thoroughly remove the old worn-out lime mortar in the joints of the brickwork to a depth of fully 1 in., after which the bricks were cleaned by brushing with wire brushes. The joints were then pointed up carefully with a cement composed of fireclay mixed with portland cement and water. It was found that an addition of ammonia water tended to retard the setting somewhat and render the mixture more resistant to temperature changes, the amount added being sufficient to give the mixture a fairly strong odor of ammonia. The joints were made flush with the surfaces of the bricks and allowed to set for two days.

After the cement had solidified, the whole setting, brickwork and iron were painted with a heavy, slow-drying asphaltum-base paint, of such character as to give a flexible coating not easily cracked or peeled from the bricks. It was necessary to first point up the joints with the cement-fireclay mortar because some of the cracks were larger than the paint could safely close effectively. All the door frames were set tightly in the brickwork and then given a heavy coat of the same paint, especially where the iron joined the brickwork. Three coats in all were applied, each being allowed to dry somewhat before the application of the next. The total time consumed in improving this setting was 10 days. The results produced caused us to carry out a similar program on all the other settings.

Herewith are some figures, calculated on the basis of pure carbon for fuel, showing the saving which may be obtained when boiler settings are maintained in air-tight condition. It should be noted that the conditions in this plant were not abnormal; in fact, the writer, since this particular investigation, has observed many cases far worse.

The chart in the figure shows the result of tests covering 27 days in terms of CO_2 in the gases of combustion. The samples were taken at the top of the first and at the top of the last pass in the gas travel of a 500-hp. Stirling boiler, fitted with a Roney stoker. The draft was natural, averaging 0.175 in. of water over the fire. The CO_2 tests were made simultaneously by two operators, who changed positions and Orsats frequently in order to eliminate the personal equation as far as possible.

It is interesting to note the regularity with which, for the most part, fluctuations in CO_2 in the first pass are followed by fluctuations in the last.

The average CO_2 at the top of the first pass was 13 per cent. for 27 days before the setting was overhauled and 8.8 per cent. at the top of the last, a difference of 4.2 per cent. This loss in CO_2 is undoubtedly a measure of the dilution of the gases of combustion due to air of infiltration and corresponds to an increase in the excess air equal to 75 per cent. of that theoretically required. The excess at the top of the first pass was close to 60 per cent. and 135 per cent. at the top of the last. This increase in excess air (75 per cent.) corresponds to a loss in fuel of approximately 5.5 per cent. (based on pure carbon).

After rendering the setting practically air-tight, the difference in CO_2 content of the gases was reduced to 0.5 per cent., the CO_2 at the top of the first pass being 13.6 per cent., as against 13.1 per cent. at the top of the last. This difference represents closely an increase in excess air of only 5 per cent. (compared with an increase of 75 per cent. formerly), which in turn approximates a fuel loss of only 0.25 per cent., or practically zero when the

accuracy of analysis of gases and unavoidable error are considered.

There can be no doubt of the value of careful treatment of boiler settings; the figures are strong evidence that all heat-absorbing parts of boilers should be housed in air-tight settings, there being considerable value in the suggestion that all boiler brickwork be covered with sheet iron, applied in sections so that working parts may be accessible for cleaning, etc. Excellent results may be obtained, however, with suitable paint. Such a paint is gummy in nature, heals when scratched and gives a heavy, impervious coating which does not become "tacky" even at the front part of the setting near the combustion chamber. There are several such paints on the market; the paint we used is sold in all the principal cities of this country.

CO_2 ANALYSIS IN PER CENT.

Boiler Setting	Before		After	
	First Pass	Last Pass	First Pass	Last Pass
A	10.7	7.2	10.9	10.3
B	15.0	10.1	14.8	14.7
C	11.85	7.3	12.1	11.3
D	13.6	7.6	13.22	12.4

EXCESS AIR CORRESPONDING TO ABOVE CO_2 CONTENTS

A	93.0	187.5	90.0	100
B	40.0	105.0	41.0	40.85
C	73.0	185.0	71.0	82.5
D	52.5	172.5	56.0	67.5

FUEL LOSS CORRESPONDING TO DIFFERENCE IN CO_2 BETWEEN FIRST AND LAST PASS, PER CENT.

A	8.25	0.75
B	5.75	0.00
C	10.00	1.00
D	10.75	0.90

Above are figures showing the results of such treatment on several other settings figured as above. The savings shown in fuel consumption are, of course, calculated theoretically. However, actual tests in which the fuel was weighed and calculated in terms of B.t.u. per kilowatt-hour of current generated, before and after treating the settings, showed close checks on the theoretical figures.

In all this work the same fuel, Hocking Valley slack, was used and showed a heat value of about 12,800 B.t.u. per pound, and a content 8.5 per cent. ash and 1.5 per cent. sulphur.

Calculating Weight of Stacks

For use in estimating the weights of circular steel stacks and smoke breechings, the accompanying table is given by Charles R. Courtenay in the *American Machinist* for Sept. 11, 1913. The table gives the actual weight of a round steel tube 1 ft. long, including longitudinal lap. For stacks No. 8 to No. 3 gage, it is only necessary to add $\frac{1}{12}$ or $8\frac{1}{3}$ per cent. of the weight given for every girth seam in the stack.

Example—How much will the plate of a 22-in. diameter by 20-ft. stack weigh if the material is No. 16 gage stock and the length of each section, including lap, is 50 in. From the table, the weight per foot is 16.4 lb.

$$20 \times 16.4 = 328 \text{ lb.}$$

There are four girth seams because

$$\frac{20 \text{ ft.} \times 12}{50} = 5, \text{ nearly}$$

and $5 - 1 = 4$.

Then

$$8\tfrac{1}{3} \text{ per cent. of } 16.4 \times 4 = 5.46 \text{ lb.}$$

is the extra weight for seams (girth) and

WEIGHT-CALCULATION TABLE. WEIGHT PER FOOT OF ROUND STACKS

Diameter	Use ¼" Rivets—Lap 1¼"								Use ⅜" Rivets—Lap 1"											
	0	1	2	3	4	5	6	8	10	12	14	16	18	20	22	24	26	28	30
	16″	32″	64″	4″	64″	32″	64″	64″	64″	64″	64″	16″	16″	16″	32″	16″	16″	4″	16″
8										11	7.6	6.1	4.9	3.68	3.04	2.65	1.82	1.52	1.23
9										12.4	8.55	6.8	5.5	4.02	3.51	2.73	2.05	1.71	1.34
10									17	13.8	9.4	7.8	6.06	4.53	3.78	3	2.27	1.87	1.53
11									18.7	15.	10.4	8.31	6.64	4.97	4.14	3.22	2.49	2.07	1.66
12									20.3	16.4	11.3	9.04	7.36	5.41	4.51	3.61	2.71	2.26	1.84
13									22	17.7	12.2	9.8	7.8	5.85	4.82	3.91	2.93	2.45	2.01
14									24.3	19	13.1	10.51	8.41	6.46	5.24	4.2	3.15	2.62	2.10
15									25.3	20 4	14.1	11.2	9.	6.73	5.61	4.49	3.37	2.81	2.22
16								33.7	27	21.7	15	12	9.58	7.17	5.98	4.79	3.55	3	2.49
17								35.6	28.6	23	15.9	12.7	10.2	7.51	6.34	5.08	3.74	3.18	2.5
18								37	30.3	24.4	16.8	13.4	10.8	7.96	6.71	5.37	4.03	3.36	2.64
19								39	32	25.7	17.7	14.2	11.3	8.49	7.17	5.66	4.25	3.54	2.86
20							49	41	33.6	27	18.7	15	11.9	9.2	7.44	5.95	4.47	3.73	3.0
21							51.5	43	35.2	28.4	19.6	15.6	12.5	9.47	7.81	6.25	4.69	3.91	3.15
22							53.8	45.1	36.9	29.7	20.5	16.4	13.1	9.81	8.17	6.54	4.91	4.1	3.36
23							55.9	47.1	38.5	31	21.4	17.1	13.7	10.2	8.54	6.73	5.13	4.28	3.45
24							58.7	49.1	40.2	32.4	22.3	17.8	14.3	10.6	8.9	7.12	5.41	4.46	3.56
25						65.8	61.1	51.1	41.8	33.7	23.2	18.6	15	11.1	9.27	7.42	5.57	4.65	3.68
26						68.3	63.5	53.1	43.5	35	24.2	19.3	15.5	11.5	9.64	7.71	5.79	4.82	3.86
27						71	66	55.2	45.1	36.3	25 1	20.1	16.04	12.1	10.	8	6.01	5.02	4.01
28						73.5	68.2	57.2	46.8	37.7	26	20.8	16.6	12.3	10.3	8.3	6.23	5.2	4.15
29						76	70.6	59.2	48.4	39	27	21.5	17.2	12.8	10.7	8.59	6.44	5.38	4.30
30				89.8	84.3	78.6	73	61.3	50	40.4	27.8	22.3	17.8	13.3	11.1	8.92	6.67	5.57	4.45
31				92.8	87	81.2	75.4	63.2	51.7	41.7	28.7	23	18.4	13.8	11.4	9.18	6.89	5.75	4.59
32				95.7	89.8	83.8	77.8	65.3	53.4	43	29.7	24.1	19	14.2	11.8	9.47	7.16	5.93	4.74
33				98.6	92.6	86.3	80.2	67.2	55	44.4	30.6	24.5	19.6	14.6	12.2	9.76	7.33	6.05	4.89
34				101.6	95.3	89	82.6	69.3	56.7	45.7	31.5	25.2	20.2	15.1	12.5	10	7.55	6.3	5.09
35			111	104.5	98.1	91.4	85	71.3	58.3	47	32.4	26	20.7	15 5	13	10⅓	7.77	6.42	5.16
36			114.2	107.4	100.8	94	87.3	73.3	60	48.4	33.3	26.7	21.3	16	13.3	10.6	7.92	6.6	5.33
37			117.3	110.4	103.5	96.6	89.7	75.3	61.6	49.7	34.3	27.4	21.9	16.3	13.6	10.9	8.26	6.85	5.48
38			120.4	113.3	106.2	99.1	92.3	77.4	63.3	51	35.2	28.1	22.5	16.8	14.01	11.2	8.49	7.02	5.63
39			123.5	116.3	109	101.7	94.5	78.7	65	52.3	36.1	28.9	23	17.2	14.4	11.5	8.68	7.15	5.77
40	149.2	134.1	126.6	119.2	111.8	104.3	97	81.4	66.6	53.7	37	29.6	23.7	17.7	14.8	11.8	8.89	7.39	5.92
42	156.4	140.7	132.9	125.1	117.2	109.4	101.6	85.4	70	56.4	38.8	31.1	24.9	18.5	15.5	12.3	9.34	7.76	6.14
44	163.7	147.3	139.1	131	122.8	114.6	106.4	89.5	73.2	59	40.7	32.5	26.03	19.4	16.2	12.9	9.75	8.07	6.5
46	171	153.9	145.4	136.8	128.3	119.6	111.2	93.5	76.5	61.7	43.5	34	27.2	20.3	16.9	13.5	10.1	8.45	6.8
48	178.4	160.5	151.6	142.7	133.8	125	116	97.6	79.8	64.3	44.3	36.1	28.4	21.2	17.6	14.2	10.6	8.82	7.09
50	185.7	167.2	157.9	148.6	139.3	130	120.7	101.6	83	67	46.2	37	29.6	22.1	18.4	14.6	11	9.18	7.39
52	193.1	173.7	164.1	154.4	144.8	135.6	125.5	105.6	86.4	69.7	48	38.4	30,7	23.1	19.1	15.3	11.5	9.6	7.68
54	200.4	180.4	170.3	160.3	150.3	140.3	130.3	109.7	89.8	72.4	50	40	31.9	23.8	19.9	15.9	11.9	10	7.97
56	207.8	187	176.6	166.2	155.8	145.5	135.1	113.7	93.5	75	51.2	41.4	33.1	24.7	20.6	16.5	12.3	10.3	8.27
58	215.1	193.6	183	172.1	161.4	150.6	139.8	117.6	96.5	77.7	53.5	42.6	34.3	25.6	21.3	17.1	12.8	10.7	8.56
60	222.4	200.2	189.1	178	167	155.7	144.6	121.8	99.7	80.3	55.4	44.3	35.4	26.5	22	17.6	13.2	11	8.85
62	229.8	206.8	195.3	183.8	172.4	160.8	149.4	125.8	103	83	57.2	45.8	36.9	27.4	22.8	18.2	13.7	11.4	9.15
64	237.1	213.4	201.6	189.7	178	166	154.2	130	106.3	85.7	59	47.2	37.8	28.2	23.5	18.8	14.1	11.8	9.4
66	244.5	220	207.8	195.6	183.4	171.2	159	134	109.6	88.3	60.3	48.7	39.3	29.2	24.2	19.4	14.6	12:1	9.7
68	251.8	226.6	214	201.4	189	176.4	163.7	138	112.9	91	62.7	50.2	40.3	30	25	20	15	12.5	10
70	259.1	232.2	220.3	207.3	194.3	181.4	168.5	142	116.2	93.6	64.5	51.6	41.4	31.1	25.7	20.6	15.4	13	10.2
72	266.5	238.8	226.5	213.2	200.5	186.8	173.3	146	119.5	96.3	60.4	53.1	42.7	31.8	26.5	21.2	16	13.4	10.6

Add 15½% of above for each girth seam. Add 8⅓% of above for each girth seam.

These weights include allowances for over weights as adopted by The Ass'n American Steel Mf'rs. Does not include bands, guy wire and rivets.

$$328 + 5.46 = 333.46 \text{ lb.}$$

is the total weight of plate in the stack.

If the gage stock had been No. 0 to No. 6, we would have had to use 15⅝ per cent. instead of 8⅓ per cent., as the lap for these gages is 1⅞ in. instead of 1 in. for the lighter-gage stock. To obtain the total weight of material in a stack, it is only necessary to add to the plate weight the weight of rivets and bands.

Economical Power-Plant Combination*

By W. J. Smith

SYNOPSIS—Steam boilers are used to generate steam for manufacturing purposes, but power is obtained from a gas-producer, gas-engine plant. An interesting arrangement is that of using the ammonia condenser water in the gas-engine jackets, for cooling the engine-exhaust pipe and in the producer humidifier and lastly pumping the remainder to the boilers.

Under certain conditions combinations of prime movers may be made which will produce economy and reliability of operation. An interesting example of this is found at the plant of the Olympia Brewing Co., Olympia, Wash., where a combination of producer-gas engine and oil-fired steam boilers has brought about remarkably good economy and efficiency. Apparently, due to certain requirements of the industry, no other form of power, as at present developed, can successfully displace that in use.

Boilers

Both boilers are water-tube, each of 250 hp. capacity. The steam requirements are such that only one is in service at a time, and they are used alternately, a month at a time, which serves to keep both in better condition than if one were idle all the time. The steam is used in the various departments for heating and brewing requirements, none being used for motive power. Both boilers are operated at high pressure, reducing valves being placed in the steam lines as required. The returns are handled by tilting traps on the various lines, which in turn deliver to a direct-return tilting trap, mounted above the boilers.

The gas producer was developed and practically built on the premises, and is designed primarily for the low-grade lignite coals of this region. It is of 250 hp. capacity, and is charged from an overhead hopper or coal pocket, the normal charge being about 850 lb., which is introduced through four feed openings or hoppers, every three hours. The gas product is used in a 250-hp. gas engine, belted to a 125-ton ice machine, and a 150-kw., 220-volt, direct-current generator, used for lighting and motors on the numerous pumps and auxiliaries about the plant.

Water Supply

An ice machine serves the various cooling processes in the brewery, and also furnishes 25 tons of ice per day, which is used for icing cars, city ice service, etc. Water used for cooling the ammonia condensers is brought to a nearly uniform temperature of 68 deg. F.; 30 per cent. of this is used in the jackets of the gas-engine cylinders, which brings it up to a temperature of 130 to 140 deg. F. From the cylinder jackets the water is led to a jacket about the exhaust pipe of the engine and an exhaust-heat economizer which raises it to nearly boiling point and at the same time cools the burned gases of the exhaust and makes the discharge of the latter much less violent and noisy. From this point the water is piped at its now high temperature to the humidifier part of the producer apparatus, where the entering air is given the proper moisture content, making a gas in which the various necessary elements are combined. This last stage reduces the water temperature to between 160 and 170 deg. F. Part of it is used for boiler-feed water and before entering the boiler is again brought to a high temperature by circulating through a coil in the last pass of the waste gases before the latter enter the stack. The remaining water is then piped over the plant and is used in all washing processes, such as for beer packages, etc., its temperature being from 20 to 30 deg. above the pasteurizing point, which is 140 deg. F.

All of the foregoing has been accomplished through what are common avenues of extensive loss; first, the necessary cooling of the ammonia condensers; second, the essential cooling of the gas-engine cylinders and exhaust gases, and, third, the waste heat in the flue gases of the boiler furnace.

An engineer and a fireman operate the power plant, the latter taking care of both boilers and the producers. The boilers being oil-fired, the feed water handled by electrically driven pumps and the return system of traps feeding the boiler, the plant is more or less automatic in operation. The producers require attention every three hours and it takes 20 min. to clean and charge at each period.

Fuel

Locally mined coal is used, costing $1.70 per ton delivered, only the lower grades being used, in fact, a considerable quantity of it was formerly waste. Although the freight haul is short, the transportation item is nearly 50 per cent. of the cost of the fuel. About 7200 lb. is used daily. At times an overload of from 10 to 20 per cent. is carried by the gas engine, nevertheless, extended tests have shown the fuel consumption to be in the neighborhood of 1.7 lb. per hp.-hr.

The average analysis of the coal used is as follows:

	Per Cent.
Moisture	16.2
Volatile matter	28.9
Fixed carbon and sulphur	32.1
Ash	22.8
Total	100.0
Heat value as fired, B.t.u.	7560
Heat value dry coal, B.t.u.	9021

For steam generation about 12 bbl. of California crude oil is used daily, costing $0.95 per bbl., delivered at the

*The operating conditions of the gas-engine part of this plant were described in detail in "Power," Apr. 29, 1913, p. 604.

plant. With this oil is mixed a tar byproduct which is obtained from the gas scrubbers. This tar, amounting to 150 lb. daily, is rich in hydrocarbons and high in heat value. For a considerable period its use was a problem and it went to waste. Later, however, by a system of reheating to render it more fluid and to better separate the water content, it was found to mix perfectly and thus was developed one factor in making the plant highly economical.

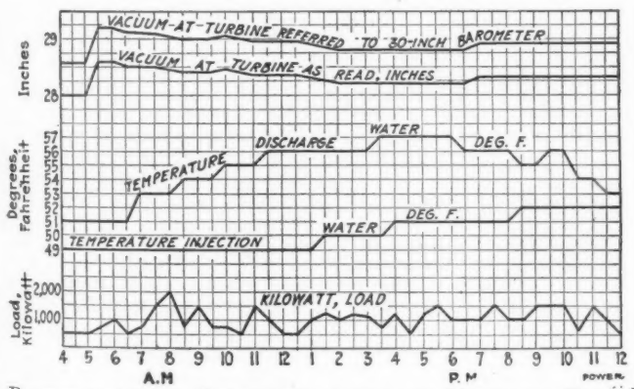
For the gas engine about one gallon of lubricating oil is required daily, costing $0.35 a gallon. This seemingly small quantity is found ample and after three and a half years' constant service the cylinders are in apparently as good condition as when installed.

The fuel cost for operating a plant of this capacity is extremely low. Comparison of operating costs between the plant as formerly driven entirely by steam power with simple noncondensing Corliss engines, and as at present equipped, shows a saving averaging more than $20 per day—the daily consumption of fuel oil alone then being about 45 bbl. It is likely that the actual saving is even greater than this, as additional equipment, requiring much more power and steam consumption, has been added since the installation of the present power plant.

Operating Condenser without Hotwell Pump

By H. C. Carroll

During the floods in March of this year, it became necessary to operate a large condensing turbo-generating unit without its motor-driven hotwell pump, the motor of



Performance of Condenser without Hotwell Pump

which had become water-soaked. This, necessarily, meant that the unit would have to be operated noncondensing, but with only a bunker full of coal and over a 1000 ft. of the railroad siding washed out, it was evident that the coal in the bunker would not last until coal could again be brought in the regular way.

The generating unit consisted of a 6000-kw., double-flow turbine direct-connected to a 10,000-kv.-a., 25-cycle generator. The auxiliary apparatus consisted of an 18,000 sq.ft. dry tube surface condenser, engine-driven circulating pump, steam-driven air pump and the motor-driven two-stage hotwell pump.

It was decided to try operating condensing, letting the air pump take care of both the noncondensible vapors and the condensed steam as well. How well this did so, is shown by the accompanying set of curves derived from the half-hourly readings taken from the daily log sheet. How much more condensed steam the air pump could handle, is not known, but it handled all the condensed steam from the largest load at that time, up to 2000 kw. The bunker full of coal lasted nearly ten days when coal was again available over the siding.

A Decimal Classification for Power-Plant Data

By Frank H. Jones

Assuming the integral number 10 as representing a typical power plant, drawings and other data relating to design and actual work of construction may be conveniently classified and marked for filing as follows:

10. POWER PLANTS

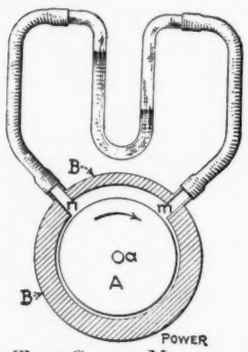
10.0 GENERAL
10.001 Studies and schemes.
10.01 Preliminary to Construction (Information necessary for a clear interpretation of the existing conditions on the site at the time of contract).
10.011 General Plan and Location, Topography.
10.012 Surface and Subsurface Investigations.
10.02 Canvass, Progress, Completion, Maintenance.
10.021 Bidders and Canvass or Bidders.
10.022 Progress Data.
10.023 Completion (guarantee of workmanship and material).
10.029 Maintenance, Instructions for, general.
10.03 Contractors Force, Material, Plant, Inspection.
10.31 Contractors Plant.
10.032 Force Accounts, Labor.
10.033 Material, Tests.
10.039 Engineering and Inspection.
10.04 Railroad and Highway Connections.
10.041 Foreign Trackwork.
10.042 Highway Connections.
10.1 SUBSTRUCTUkE
10.11 Excavation, Filling, Surfacing, Grading.
10.12 Retaining Walls, Quay Walls.
10.13 Piles and Pile Records.
10.2 SUPERSTRUCTURE (including coaling and ash handling facilities and other minor structures).
10.21 Floors, Walls, Elevations, Sections, Roof.
10.211 Foundations.
10.22 Structural Steel.
10.3 PIPING
10.31 Underground Piping,
10.32 Plumbing, Commercial Water Supply.
10.33 Fire Protection System.
10.34 Plumbing, Sewage and Drainage.
10.35 Heating and Ventilating.
10.36 Other Pressure Systems.
10.39 Pipeway Construction, Ducts, Manholes, Dry Wells, Pits.
10.4 ELECTRIC POWER AND LIGHT (for the structure itself).
If outside service is used the wiring data up to the main switchboard is classified herein (10.4).
If an isolated plant is installed for this purpose the construction and wiring data shall be classified herein (10.4).
10.41 Distribution (includes all wiring layouts from, and including, the main switchboard to the finished outlet.
10.42 Appliances (Details from the point of outlet).
10.5 CHIMNEY
10.51 Foundation.
10.52 Stack.
10.6 POWER-GENERATING EQUIPMENT
10.61 Boilers.
10.611 Settings, Smoke Flue.
10.612 Fittings and Apparatus.
10.62 Steam Engines.
10.621 Reciprocating.
10.622 Turbine.
10.63 Hydraulic Engines and Machines.
10.631 Waterwheels.
10.632 Turbines.
10.633 Pumps.
10.639 Miscellaneous.
10.64 Electrical Generation and Distribution.
10.641 Generators and Motors.
10.642 Storage.
10.643 Transformation.
10.644 Transmission.

The absence of strictly technical classification under some of the headings, notably 10.6, is a commendable feature where the system is to be handled by index clerks with little or no technical knowledge. Though draftsmen and engineers would be able to use a more detailed classification than is shown, the majority of general office clerks would not be able to apply such a thorough outline with accuracy and intelligence.

The logical arrangement of data and the inclusive divisioning eliminate the need of an alphabetically arranged working key and as a standard classification can be readily adapted to large, small and general power-plant engineering data.

Dr. Gaede's Molecular Air Pump

The antithesis of the "viscosity" turbines of Tesla and others, which turbines have no blades but are carried around by the fluid friction of the steam, is the molecular air pump of Doctor Gaede. This consists simply of a cylinder *A* mounted upon a shaft *a* and running freely but fairly closely within the casing *B*. A pocket *nm* is formed in the casing and as the cylinder is revolved the air is carried with it, producing a difference in pressure at the different ends of the pocket, as shown by the manometer in the sketch. The inventor says that with a pump driven by a ⅓-hp. motor a Roentgen tube of 1 liter volume can be exhausted to a vacuum of 5 mm. in 10 sec. The operation is from five to ten times as rapid as in the ordinary pump.



THE GAEDE MOLECULAR AIR PUMP

The following table gives the relations between the pressure differences and the revolutions per minute, but our information, which comes from *Die Naturwissenschaften,* does not include the diameter of the runner nor allow the peripheral speed to be calculated:

r.p.m.	p_1	p_2
12,000	0.05	0.0000002
12,000	1.00	0.000005
12,000	10.00	0.00003
12,000	20.00	0.0003
6,000	0.05	0.00002
2,500	0.05	0.0003

OBITUARY

JAMES T. HADLEY

James T. Hadley, aged 42, secretary-treasurer of the Reid Gas Engine Co. and of the Reid Land Development Co., died at his home in Oil City, Penn., Dec. 28, after a long illness.

MURRAY FORBES

Murray Forbes, aged 50 years, died at his home in Greensburg, Penn., Dec. 28. He was a native of Philadelphia. In 1886 he settled in Greensburg and since then had been identified prominently in the water plant and power business. At the time of his death he was treasurer and general manager of the Westmoreland Water Co. He was a heavy stockholder in the Derry Water Co., Irwin Water Co. and the Dennison Water Co., Dennison, Ohio. Mr. Forbes was a son of William S. Forbes, professor of surgery at Jefferson Medical College, Philadelphia, and was a member of several scientific and engineering societies.

HERMAN MEINHOLTZ

Herman Meinholtz, vice-president and superintendent of the Heine Safety Boiler Co., died Dec. 24. His death was due to heart trouble, following an illness of two weeks, and occurred at his home in St. Louis. He was born in St. Louis, Feb. 7, 1868, and received his education in the St. Louis grammar schools and the manual training school and night classes of Washington University. Following a year of work with Prof. J. B. Johnson, at Washington University, Mr. Meinholtz was successively employed with Shickle, Harrison & Howard, the N. O. Nelson Manufacturing Co., and the Heine Safety Boiler Co., all in St. Louis. From 1895 to 1900 he was engaged in supervising boiler construction in various contracting shops. Since that time he has been in full charge of the Heine shops. The present shops, completed in 1909, were designed and their construction supervised by him. Mr. Meinholtz was a member of the American Society of Mechanical Engineers and an active participant in the work of the Boiler Rules Committee.

Los Angeles Power Plans

Plans to develop a municipal power project, involving an expenditure of $10,000,000 in Mono County, were outlined before the State Board of Water Power Control by attorneys and engineers representing the city of Los Angeles recently.

A tunnel 26 miles long, to carry water from streams tributary to Mono Lake into the Owens River valley, where it will be merged with the Loss Angeles aqueduct project, is one of the most pretentious features of the proposed undertaking.

Three power plants in Mono County are also projected. These plans were revealed in explanation of Los Angeles' opposition to the application of E. G. Ryan, a Southern Pacific detective, who had asked for permits to appropriate water from Rush Creek and Leeving Creek, in the vicinity of Mono Lake. At the conference Assistant City Attorney S. B. Robinson declared that it was his opinion that Ryan was representing the Chappell interests, which control the Nevada-California Power Co., the Sierra Power Co., the Mono Power Co., and the Hydro-Electric Co. Ryan's attorneys deny this allegation but have refused to make their financial backing public.

PERSONALS

Walter L. Mulligan, manager of the United Electric Light Co., Springfield, Mass., was presented a large silver loving cup by the members of the United Electric Light Co.'s Employees' Relief Association on Dec. 19.

W. J. Kerr, for over three years chief draftsman for the Green Engineering Co., has become associated with W. A. Bloneck & Co., manufacturers of boiler efficiency meters, and will have his headquarters at the main offices in the Fisher Building, Chicago, Ill.

Norway's Electrical Production—In Norway 12 years ago there was practically no production of electricity worth mentioning, but the horsepower now generated or being developed is approaching half a million. The practice has been to harness existing waterfalls or to dam natural lakes. Johan Haare, of Trondhjem, has worked out a completely new project which consists in the construction of an artificial lake as a reservoir for a small river, the Surna, in Nordmöre. The course of the river is to be altered by leading the water through a tunnel in a mountain a distance of two miles, thus obtaining a considerable fall. The electrical power station will be located at the lower end of this tunnel. About 50,000 hp. will be made available.

The Oregon Hydro-Electric Commission has completed its organization here and outlined a plan of procedure in its work which includes a study of every phase of production and utilization of electric power in the Northwest. The commission plans to make investigations in all counties where electric power is being developed from water power. It intends to urge Washington and Idaho to organize similar commissions and coöperate in the work, so that the problems of water-power development and application of electricity to manufacturing and other power purposes may be worked out. One of the largest projects to be investigated under the new commission is the canalization of the Snake and Columbia rivers, for improving navigation and developing hydro-electric power in large quantities. Idaho and Washington will be asked to coöperate in this survey, and the effort aimed at is to interest the Federal Government and devise methods whereby, along the rivers, districts may be organized to carry out the proposed power and navigation projects.

Duty to Safeguard Electric Wires—A company which maintains wires carrying deadly currents of electricity must use the highest degree of care to avoid injury to persons who, in their lawful employment, are liable to come in contact with them. "One who is brought by his employment in close proximity to electric wires, which are apparently insulated cannot be fairly charged with contributory negligence in coming in contact with the wires, unless the contact was the result of heedlessness or of his own lack of proper precautions for his safety." (Pennsylvania Supreme Court, Yeager vs. Edison Electric Co., 88 Atlantic Reporter 872). An electric company whose wires were strung over a highway was bound to anticipate injury to one while driving a hay derrick along the highway, on account of the derrick coming in contact with an uninsulated wire, if it was previously a common practice for persons in the locality to drive derricks along public road. (Oregon Supreme Court, Greenwood vs. Eastern Oregon Light & Power Co., 136 Pacific Reporter 336).